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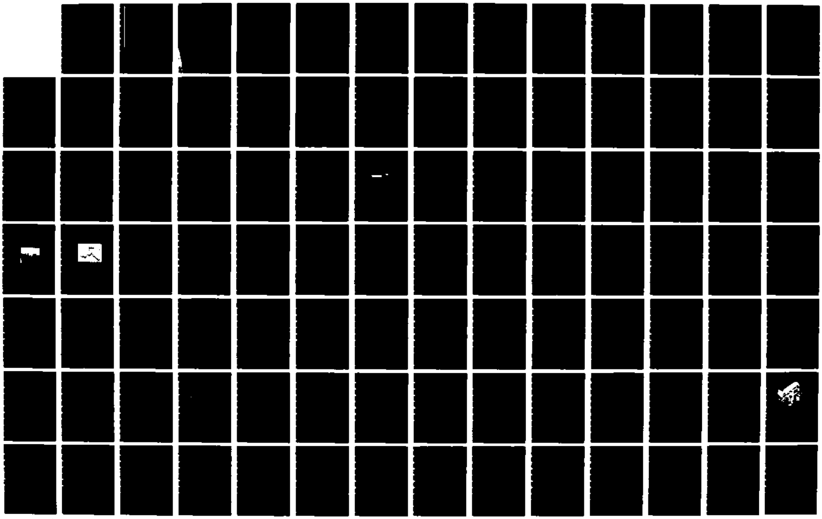

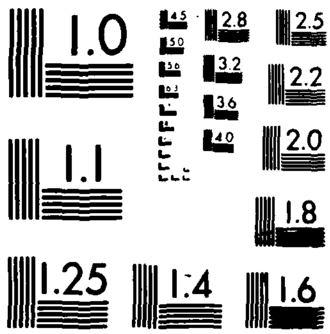

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Report No. A5/84

FINAL REPORT

DETECTION OF DAMAGE IN HYDRAULIC COMPONENTS

BY

ACOUSTIC EMISSION TECHNIQUES

FOR

U.S. ARMY MOBILITY EQUIPMENT RESEARCH AND

DEVELOPMENT CENTER

Fort Belvoir, Virginia 22060

CONTRACT NUMBER

DAAK70-82-C-0104

APRIL, 1984

PREPARED BY

FLUID POWER RESEARCH CENTER
OKLAHOMA STATE UNIVERSITY
STILLWATER, OKLAHOMA 74078

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Acoustic Emission Instrumentation was applied to the detection of damage, wear and leakage in hydraulic pumps, valves and cylinders. The results show that these components emit signals between 20 and 90 KHz that can be correlated to leakage or damage. A literature survey and a summary of commercially available instruments are also included.

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NOMENCLATURE

A.E.	Acoustic Emission
AVLD	Acoustic Valve Leak Detector
dB	Decibel
F	Frequency (Hertz)
FFT	Fast Fourier Transform
Hz	Hertz (Hz)
KHz	Kilo Hertz (1000 cycles/second)
NB	Narrow Band
RPM	Revolutions Per Minute
RMS	Root Mean Square
psig	Pounds Per Square Inch (Gage)

CHAPTER I

INTRODUCTION

The goal of this study was to determine whether "acoustic emission" type instrumentation could be used to detect damage in hydraulic components. The basic premise of this work was that worn or damaged pumps, cylinders, and valves probably emit high frequency stress waves that can be readily detected by piezoelectric transducers at frequencies greater than 20 KHz.

The following work describes the various tests conducted on commercial hydraulic components, the relevant literature, and information about commercially available instrumentation.

Various types of pumps, valves and cylinders were tested on special test stands. Tests were also run on an operational front loader to gain data under field conditions.

CHAPTER II

LITERATURE SURVEY

2.1 Principles and Background

The term acoustic emission refers to the so-called shock pulse which is emitted by material under dynamic stress. The pulse is generated in the stressed material and is transmitted into adjacent structures. The pulse has a very high energy content for a short duration. The frequency of the signal is well above the normal machinery vibration range, [1].

Detection of the pulse is achieved with the use of specialized piezoelectric transducers and electronic signal conditioners. After signal conditioning or amplification, the electrical signals are available at high strength. At this point, there are several processing techniques which can be used to analyze the signal. These techniques will be discussed later.

Until about 1973, acoustic emission technology was primarily used in the non-destructive testing of such structures as pipelines, heat exchangers, storage tanks, pressure vessels, and coolant circuits of nuclear reactor plants. However, the possible application of this technique to the detection of defects in the bearings of rotating equipment was recognized, [2]. This led to work in applying acoustic emission techniques to machinery surveillance systems.

Most machinery surveillance systems attempt in some way to provide the earliest possible indication of impending equipment failure. The detection of impending failure at an early stage allows action to be taken, which can either prevent the failure from occurring or allow scheduled replacement of the affected part or component on a convenience basis.

High-frequency acoustic emission technology has been shown by Drago [3] to be more effective as an early warning indicator than the popularly used low-frequency vibration and sound techniques. In fact, high-frequency technology has become commonly referred to as incipient failure detection, IFD, to emphasize this ability to detect failure in its early stages.

The basic premise of high frequency acoustic IFD monitoring is that the presence of defects in machinery and mechanical structures is characterized by corresponding abnormalities and changes in the acoustic signature. For early identification of failure, these defects must be detected when they first develop and are quite small. The amount of detectable energy released from a small defect, however, is usually negligible in comparison to normal machinery operating noise. As stated previously, though, normal machinery noise often occurs at lower frequencies, while stress wave emission caused by defects extends to much higher frequencies. It is this frequency separation that accounts for the success of IFD technology, [4].

Incipient failure information is obtained from the acoustic signature by the monitoring of certain parameters derived from the electronic signal of the acoustic emission transducer. Three commonly used parameters for IFD monitoring are the RMS, or root-mean-square parameter; SAT, or signal above threshold parameter; and PUL, or pulse count parameter. In addition, spectral analysis of the acoustic signal is used to determine incipient failure information. Further discussion of each one of these parameters follows.

The numerical value of the RMS parameter is a measure of the total effective amplitude of the measured signal across the frequency band of the IFD system. This is generally equivalent to the "smoothness" of mechanical contact, since the energy emitted is examined at a frequency band high enough to discount the effects of rotational energy.

The SAT parameter represents the amount of signal that exceeds a certain threshold value. Whereas the RMS parameter is mostly a measure of random noise, the SAT parameter is a measurement of abnormal signals of high amplitude that have a specific cause: the sudden release of energy due to mechanical impact or fracture.

The PUL parameter is used for monitoring piping and vessels. Crack propagation in these structures produces signal bursts of high amplitude. The PUL parameter is the number of these bursts counted in a given time interval. This parameter is, thus, a direct indication of the growth of cracks. The PUL parameter is not used in rotating equipment monitoring, where it has been replaced by the SAT measurement.

Frequency spectral analysis for IFD systems is obtained by implementing a Fast Fourier Transform (FFT) of the measured signal. (See Appendix E). The use of spectral analysis in IFD systems is highly significant because it can be used to isolate the frequency at which a given defect manifests itself. The amplitude increase at that frequency is a direct measure of defect progression, while the frequency itself provides clues as to the root cause or origin of the defect signal, [2].

2.2 Experimental Work

H. L. Balderston [5] of the Boeing Company examined the possibility of incipient failure detection in ball bearings. He tested ball bearings for various types of created failures. The types of failures created were improper lubrication, fatigue of ball, fatigue of outer race, worn cage, and fatigue of inner race. Using accelerometer transducers, a tape recorder, and a power spectrum analyzer, Balderston was able to record the acoustic signature data and obtained the power spectral density plots for each type of failure. The plots indicated three basic sources of noise in bearings: (1) the noise of rotation, (2) resonant frequencies of the bearing components and the transducer itself, and (3) the acoustic emission phenomenon.

Rotational noise accounted for frequencies up to approximately 2000 Hz. These noises originate from such things as a rough spot on the inner or outer race or a defect on a ball. These frequencies can be computed from knowledge of the bearing geometry and the rotational speed of the shaft. For example, the frequency of balls passing over a spot on a stationary outer race is given by the formula:

$$f = \frac{n}{2} (N) \left(1 - \frac{Dw}{dm} \cos \alpha\right)$$

where, Dw = ball diameter
dm = pitch circle diameter
α = contact angle
n = number of balls
N = speed of rotation in revolutions per sec.

Rotational frequencies are very important diagnostic tools for use in relatively simple mechanisms. However, in more complex equipment, the bearing rotational noises tend to be of a smaller magnitude than the noises produced by a major element such as a rotor. This makes the use of rotational noise difficult to use as a prediction of incipient failure.

Balderston determined resonant frequencies of the bearing components and the transducer itself to be a very good indicator of incipient failure. The resonant frequencies observed were in the range of 10 KHz to 90 KHz, which is above the range of normal machinery vibration.

Resonant frequencies are a function of the mass configuration and type of material. An expression for the lowest natural (resonant) frequency of "spheroidal vibrations" of a ball was developed by H. Lamb and described by Love, [6].

$$f = \frac{0.424}{r}\sqrt{\frac{E}{2\delta}}$$

where, r = radius of ball
E = Modulus of elasticity
δ = density of ball

The following equation describes the frequencies of free vibration of a race.

$$f = \frac{K(K^2 - 1)}{2\sqrt{K^2 + 1}} \frac{1}{a^2} \sqrt{\frac{EI}{M}}$$

where, K = order of resonance
a = radius to neutral axis
I = moment of inertia of cross-sectional area about the neutral axis
m = mass of race per linear inch

The two equations given are for "free" boundary conditions. The actual values would be slightly different in the assembled bearing.

The resonant frequencies are usually initiated by shock excitation associated with minor structural irregularities and/or defects. Balderston determined the acoustic energy at resonant frequencies were much higher than sounds associated with background noise and the sounds of rotation. Balderston also determined that the frequencies of resonance do not vary with bearing speed. Their amplitudes, however, do vary directly with bearing rotational frequency and severity of damage. These two facts are very significant in that incipient failure information can be obtained from monitoring the amplitude of the signal at these frequencies. The measured amplitude at a given speed is a direct indication of the severity of damage. In all the bearing failures Balderston created, the amplitudes of the signal at the bearing resonant frequencies yielded the incipient failure information desired.

Balderston observed acoustic emission in the bearing tests he performed; however, he did not extensively document this because the resonant frequencies proved to be adequate to detect the defects covered. He did state that, in heavily loaded bearings and in bearings operated under high temperature, the acoustic emission phenomenon becomes more important.

H. P. Block [7, 8] applied many of the principles of incipient failure detection and the work of Balderston to use in a computerized IFD system for an entire plant. The plant in question was the Exxon Chemical plant in Baytown, Texas.

Acoustic emission transducers were placed on or near bearings on many different types of machinery throughout the chemical plant. Bearings in compressors, pumps, motors, gear boxes, turbines, and extruders were the main targets of observations.

The RMS parameter and the SAT parameter for selected frequency bands were monitored with a computer. The frequency bands monitored corresponded to the resonant frequencies of the bearings. Threshold levels for each parameter and piece of equipment were stored in memory and used for comparison with readings taken periodically from the sensors. If any threshold level was exceeded, an alarm was activated, thus informing personnel of the possibility of failure.

In a 31-week period, 37 actual or potential failure events were analyzed. Early warning of incipient failure resulted in lower repair costs, fewer production losses, and decreased fire risks.

Some examples of the failure events analyzed are:

1. Bearing failure in an extruder pelletizer
2. Gear tooth damage in a gear box
3. Extruder shaft rubbing
4. Pump cavitation
5. Seal leak detected in centrifugal pump

Many other failure events were documented by Block and Finely, [9]. The most encouraging result of their work was not only could incipient failure in bearings be detected, but also pump cavitation and seal leakage. In addition, improper process procedures such as excessive feed rate to an extruder could be detected. The resulting IFD system alarm thereby caused corrective action to be taken.

J. Dickey [10], working at the David W. Taylor Naval Ship Research and Development Center, extended the use of acoustic emission techniques to the detection and measurement of valve leaks. He tested air, steam, and hydraulic valves for various leak rates The valves tested were a 2 in. ball valve used in high pressure air systems, a 3 1/2 in. globe valve used in steam systems, and a 1 1/2 in. spool valve used in hydraulic systems. Leak rates were varied for all valves by modifying the valve seating surfaces to simulate actual defects in service. Data were obtained for each valve type by recording the averaged amplitude of the A.E. signal versus frequency for the various created leak rates. Data analysis involved plotting leak rate versus signal amplitude at a specific frequency to determine how well the two variables correlate in terms of accuracy, resolution, and repeatability.

Experimental results obtained by Dickey showed that there are detectable acoustic emission signals associated with hydraulic, steam, and high-pressure air valve leakage and that the signals increased in amplitude with increasing leakage rates. The variation of valve back pressure, system pressure, and fluid temperature was determined to cause corresponding changes in the measured A.E. signal. Dickey determined that these variables must be held constant in order to obtain a usable relationship between signal amplitude and valve leak rate.

2.3 Observations from the Literature

The acoustic emission phenomenon is defined as a "generic term denoting the occurence of deformation induced sound pulses of elastic waves in metal," [5]. Two types of acoustic emissions have been characterized: (1) the "Burst" signal is an indication of material flaw growth, and (2) the so-called continuous emission due primarily to stress-induced dislocation movement. The continuous emission has also been called "friction noise" because its origin is believed to be the frictional contact between two moving surfaces, [5].

The acoustic emission signal is detected using specialized piezoelectric transducers and electronic signal conditioners. The frequencies of the A.E. signal are very high compared to those of normal machinery vibration as produced by unbalance.

The exact frequency range to use for the detection of acoustic emission is an area of disagreement among authors on the subject. Balderston [5] stated that the frequency range of the acoustic emission phenomenon is from 10 KHz to 1 MHz. Parsons [1] and Finely [2] state the range is a narrow band centered around 40 KHz. Block [7, 8] states the range most useful as the 80 KHz to 120 KHz band. This disagreement on the best range to monitor does not seem to pose a problem, however. It seems that this diagnostic range varies upon the application and is found mostly by trial-and-error. The procedure usually involves testing a piece of equipment and inflicting some type of damage upon it. The spectrum of the resulting A.E. signal then reveals the frequency range in which the defect manifests itself.

In rotating machinery, the RMS parameter and SAT parameter yield good incipient failure information. In addition, spectral analysis of the A.E. signal also provides information regarding component health. For piping and vessels, the PUL parameter yields the best incipient failure information.

As a final point of interest, transducer calibration at this time does not appear practical or necessary according to Block, [7, 8]. The present procedure is to store threshold readings of components and compare subsequent readings with the stored values to derive incipient failure information.

CHAPTER III

EQUIPMENT DESCRIPTION

The experiments performed for this project involved the use of commercially available acoustic emission equipment and transducers. The intent of this chapter is to provide a working description of the equipment for this project. In addition, practical considerations such as transducer mounting are discussed.

3.1 Acoustic Valve Leak Detector (Model 5120) [12]

The Acoustic Valve Leak Detector manufactured by Physical Acoustics Corporation is primarily designed to monitor for gas and liquid leakage in both large and small components located in a laboratory or industrial facilities. This piece of equipment was developed by Moore, Tate, and Dickey at David Taylor Naval Ship Research and Development Center in Bethesda, Maryland.

3.1.1 Principles of Operation

Sound is generated during any incipient failure, as in gas or liquid leaking on the body interface. The unit detects this leakage-associated sound through two transducers (A and B). One transducer is placed strategically to detect structure-borne noise and the other attaches to the leaky body (structure) to monitor the leakage-associated sound added to the background noise. It has been found that higher frequency sounds are attenuated to such an extent along the structure that, when electrical signals from the two transducers are subtracted, the leakage-associated electrical signal is separated from the background.

Electrical signals from each transducer are fed to preamplifiers through co-axial cables. Signals from a local oscillator and from the preamplifiers are then fed to a mixer stage via a high-pass filter, which eliminates low-frequency structure-borne noise. The mixer stage outputs, which are the difference between the amplified transducer signals and the local oscillator signal, are in the audio spectrum. This heterodyne technique allows the operator to hear a representation of the ultrasonic acoustic emissions on the earphones. The amplitude of the audio signal is also displayed on the analog acoustic amplifier meter located on the front panel. An amplitude output is provided to drive an X-Y plotter with a DC voltage proportional to the average RMS value of the input energy in a selectable band width of $\pm$ 10 KHz, with a center frequency of 10 KHz to 100 KHz. The frequency output provides a DC voltage proportional to the center frequency of the amplitude output. In the sweep mode, a voltage ramp drives the variable control local oscillator over the required 10:1 range in approximately 30 seconds.

Shown in Figure 3.1 are the front panel controls and functions and in Table 1 are the electrical specifications. For most testing only one channel was used. This channel was modified internally (Figure 3.2) and connected to the input of the Model SD 345 Spectrascope III

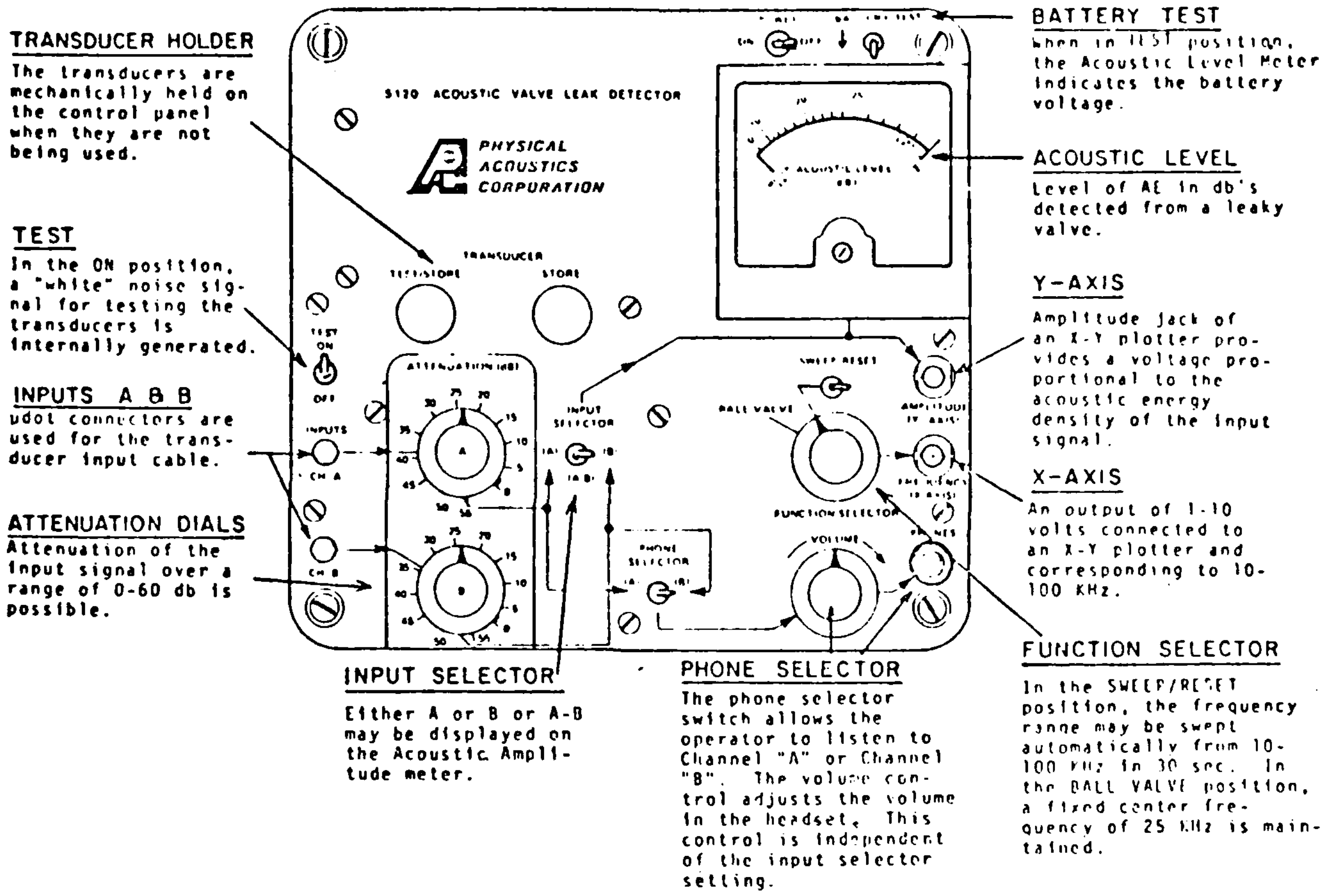


Figure 3.1: Front Panel Controls and Functions of Acoustic Valve Leak Detector [12]

NOISE: <1.5 μV RMS, 10-200 KHz BW, $Z_{IN} = 0$
< .4 μV RMS, 3 KHz BW, $Z_{IN} = 100\Omega$

GAIN: 120 DB, VARIABLE IN 5 DB STEPS

ATTENUATION: 0-55 DB IN 5 DB STEPS, ± 5%

BANDWIDTH: 10-200 KHz

RESOLUTION: .5% OR 0.4 μV RMS

DRIVE: 0-10 VDC AT 10 mA

FREQUENCY RESPONSE: ± 2 DB, 20-100 KHz BW

RANGE: 1-10 VDC ± 0.01V PROPORTIONAL TO THE CENTER FREQUENCY OF THE AMPLITUDE SIGNAL

SWEEP RANGE: 10:1, 10-100 KHz

DISTORTION: <1% TOTAL HARMONIC DISTORTION (THD) FROM 10-200 KHz

BATTERIES: ± VDC AT 60 mA FOR 8 HOURS ON A 16-HOUR CHARGE

VARIATION: ADJUSTABLE ± 12-15V AT 75 mA

INPUT IMPEDANCE: 100 Ω // 40 pF

DISTORTION: <1% AT 5.6 mV INPUT

CHANNEL SEPARATION: > 60 DB

SENSITIVITY: .25 mV RTI

PHONE OUTPUT: 3.5 MAX INTO 1KΩ

AVERAGING TIME: 0.15 ± 0.015 SEC

SENSITIVITY: ADJUSTABLE 7-10 VOLTS ± .01 VDC FOR 10 mV RMS INPUT AT 25 KHz

LINEARITY: 1%

DRIVE: 1-10 VDC AT 10 mA

DRIVE: 1V INTO 600Ω

RAMP: 10:1 RANGE IN 30 SEC ± 10% (REFERS TO THE VOLTAGE AT X-AXIS-FREQUENCY)

AC POWER: 115 VAC ± 10% 50/60 Kz
220 AVAILABLE

Table 1: Electrical Specifications of Acoustic Valve Leak Detector

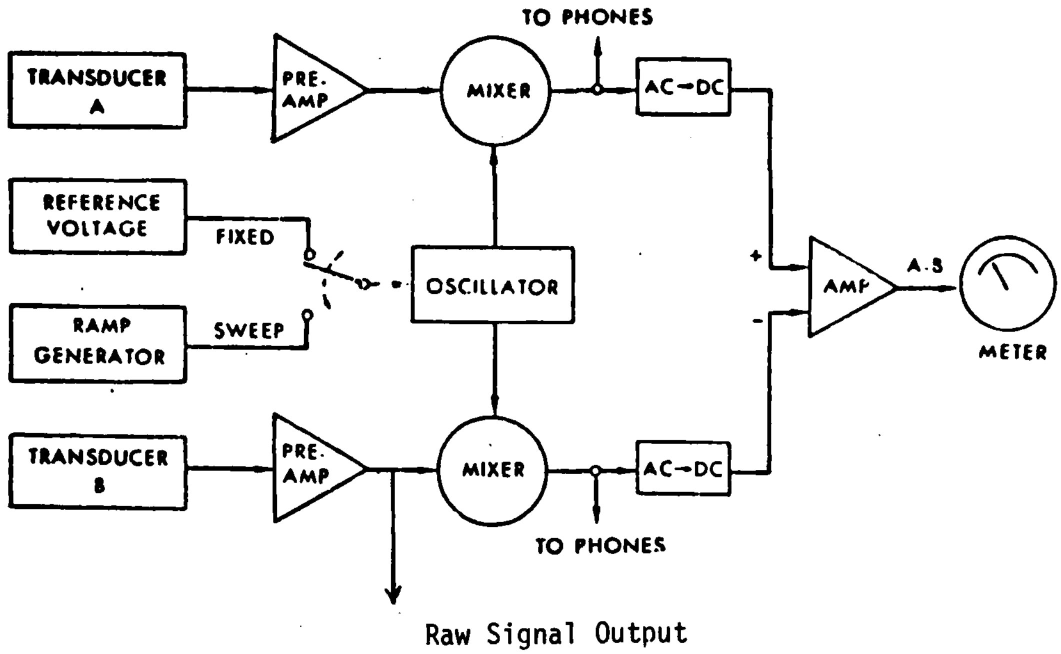


Figure 3.2: Block Diagram of Acoustic Valve Leak Detector [12]

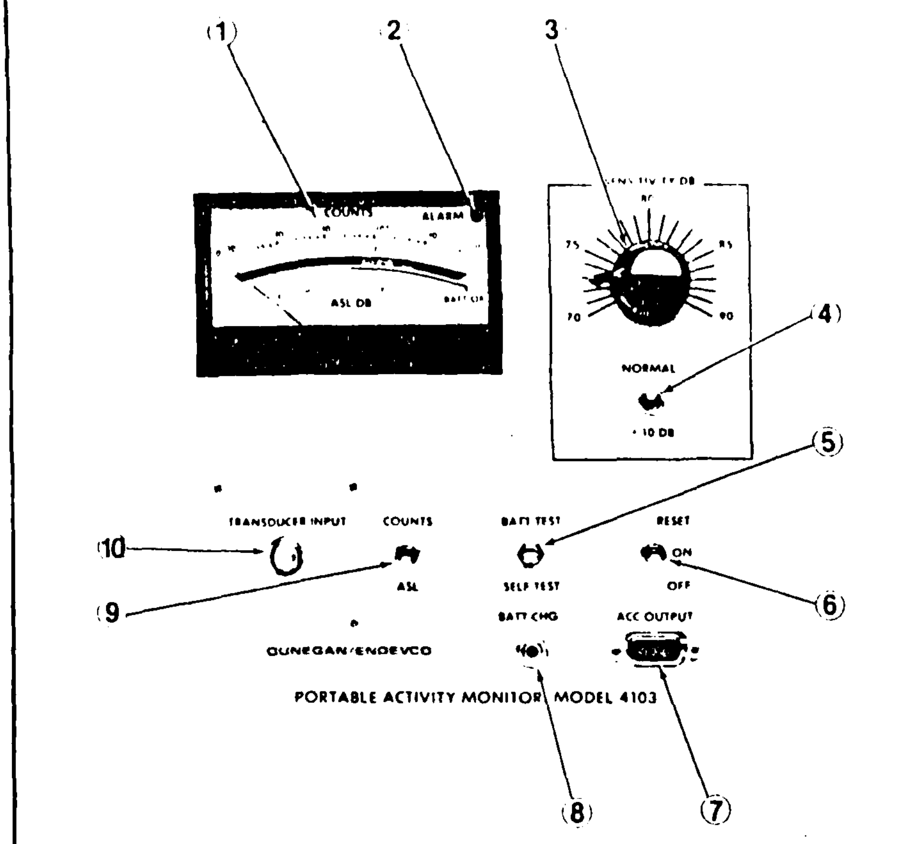
(Spectrum Analyzer) and output was obtained on an Axiom (EX-850) Video printer. This modification provided an amplified analog signal for analysis or viewing on a digital oscilloscope.

3.2 Portable Activity Monitor (Model 4103) [13]

The Dunegan/Endevco Model 4103 Portable Activity Monitor is a versatile acoustic emission (A.E.) instrument for use in industrial environments for either periodic testing or as an installed unit for semi-continuous monitoring. Characteristics that may be individually monitored are total counts, total events, count rate, event rate, or Average Signal Level (ASL). An alarm level (counts, ASL, or both counts and ASL) can be preset to indicate when a predetermined level has been exceeded. An LED on the meter face lights to indicate the "alarm" condition.

A complete system usually consists of a sensor with an interconnection cable, preamplifier, filter (switchable in or out), amplifier, digital logic circuits, and meter with logarithmic and decibel scales. When the instrument is used to monitor A.E. count activity, the Portable Activity Monitor registers "counts" detected by the sensor. Counts can be displayed as: (1) total counts or events in the totalization mode of operation, (2) count rate or event rate (four periods are available), or (3) peak count rate or peak event rate.

Frequency response for the instrument is 95-600 KHz with the high-pass filter switched in. With the filter out, the band width is 20-600 KHz. Figure 3.3 shows the front panel controls.



LEGEND:

1. DISPLAY Scale
2. ALARM Indicator
3. SENSITIVITY - DB Control Knob
4. NORMAL/+10 DB Switch
5. BATT TEST/SELF-TEST Switch
6. RESET/ON/OFF Switch
7. ACC. OUTPUT Connector
8. BATT CHG. Jack
9. COUNTS/ASL Switch
10. TRANSDUCER INPUT Connector

Figure 3.3: Front Panel Controls of Portable Activity Monitor [13]

The specification for the electrical characteristics are:

Input	Signal from differential transducer
Meter Display:	Counts logarithmic scale 0 - 10^{0} Average Signal Level (ASL dB) scale: -20 dB to + 3 dB
Gain:	70 - 90 dB, adjustable
Sensitivity Threshold Voltage:	Front panel selectable: 1 volt (normal switch position) 0.316 volt (+ 10 dB switch position)
Frequency Response	Internally selectable: 20 - 600 KHz, 95 - 500 KHz

3.3 Bearing Monitor [14]

The model 6120 bearing monitor is a portable, battery-operated instrument manufactured by Physical Acoustics Corporation designed for industrial use in ball and roller bearings.

3.3.1 Principle of Operation

The term "acoustic emission," as mentioned before, is applied to the so-called "shock pulse" which is emitted by a material under stress. In a ball or roller bearing, this pulse is generated when, for example, a crack occurs in an inner or outer race. Each time a ball or roller crosses the crack, it is stressed and a pulse is generated in the ball or the race, or both. This is further transmitted through the race into the mounting structure. The pulse is characterized by a very short rise and a very short duration and, as a result, has a very high energy content for a very short time. The frequency of the signal is well

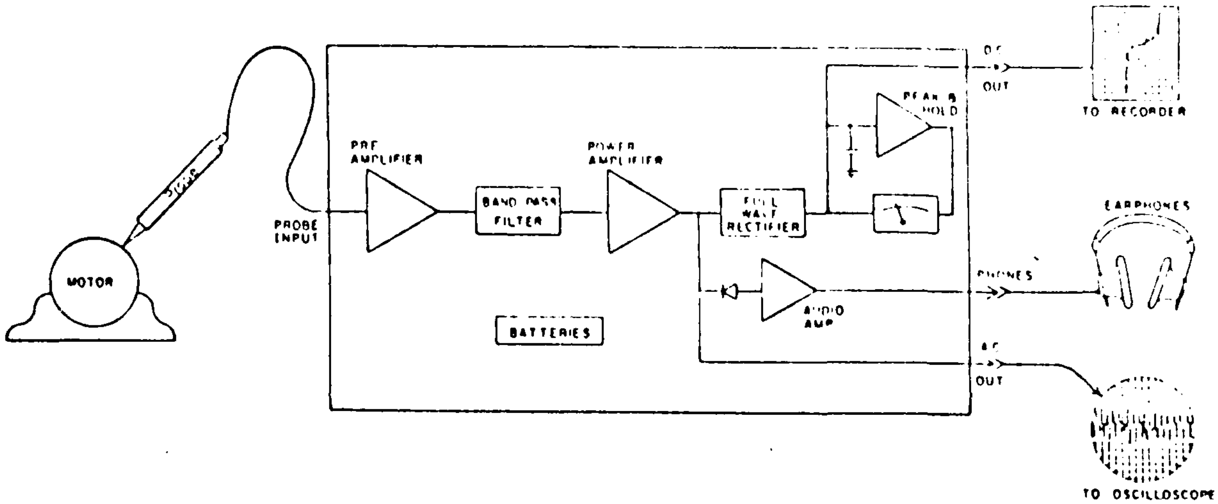


Figure 3.4: Circuit Diagram of Model 6120 Bearing Analyzer [14]

above the normal hearing range, in the band of frequencies centered about 40,000 Hz. A pulse is generated each time a ball passes over a crack and, as a result, pulses occur at a rate directly related to the number of balls and the rotational speed of the bearing.

The A.E. phenomenon is distinctly different from sounds produced as a result of vibration of the bearing or its mounting structure, since these vibrations occur at much lower rates and over a wide range of frequencies dependent on the speed of rotation, resonance of the structure, etc. Vibration frequencies range from below normal hearing (approx. 20 Hz) to 15 KHz for shaft speeds of up to 100,000 rpm. It should also be noted that acoustic emission occurs long before vibration is measurable; and, if vibration is in evidence, the bearing has long since started failing and catastrophic failure is imminent.

3.3.2 Circuitry Description of the Bearing Monitor

The measurement of acoustic emission is accomplished by detecting a 40 KHz signal using a piezoelectric crystal (refer to Figure 3.4). The crystal is mounted in a probe which can be pressed against the bearing mounting structure (or outer race, if stationary). The crystal reacts to the shock pulse and produces an electric signal which is the same frequency as the acoustic emission signal. This signal is then electronically filtered to remove all frequencies above and below the acoustic emission band (35 to 50 KHz). This removes signals due to vibration, mechanical action, and other sources not related to the bearing itself. The filtered A.E. signal is amplified to provide power needed to drive a readout system, which may be a meter or other device such as a recorder or oscilloscope. In addition, the signal may be rectified and amplified to drive earphones, so that the sound may be monitored audibly for vibrations which indicate various types of damage. Figure 3.5 is a graph showing the broad range of acoustic and vibration signals and their relation to the much higher frequency acoustic emission band.

The high frequency spectrum shown in Figure 3.6 indicates the rapidly changing A.E. signal, which is acquired by the probe and displayed on an oscilloscope. This is the frequency range from 35 to 50 KHz. The sharp spikes are shock pulses caused by the actual passage of the ball over cracks, pits in balls, foreign matter, etc. The heavy line (envelope) of the A.E. signal is shown in B to illustrate the rectified signal which is audible. The sound heard is the change in amplitude (loudness), which is heard as a soft rushing or hissing noise for a good bearing or a much louder popping or crackling noise for a bearing which has entered the failure mode. Lack of lubrication will cause sounds similar to those of a good bearing but much louder in level. The use of headphones to monitor the sound of the acoustic emission adds considerable to the basic information in determining what steps should be taken in maintenance.

The controls on the instrument along with their descriptions are shown in Figure 3.7. The important specifications are:

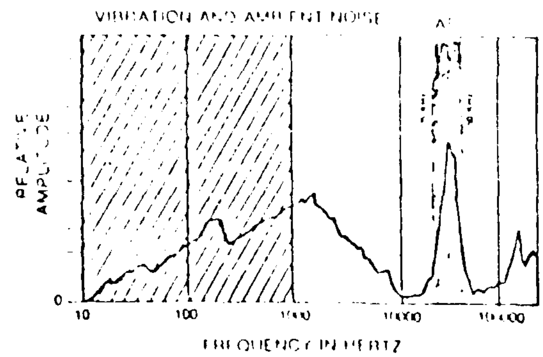


Figure 3.5: POWER SPECTRAL DISTRIBUTION OF ACOUSTIC, VIBRATION AND ACOUSTIC EMISSION SIGNALS. [14]

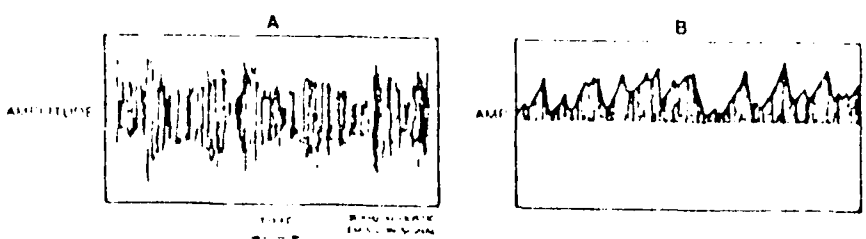


Figure 3.6: AC SIGNAL RECTIFIED (AUDIBLE) SIGNAL [14]

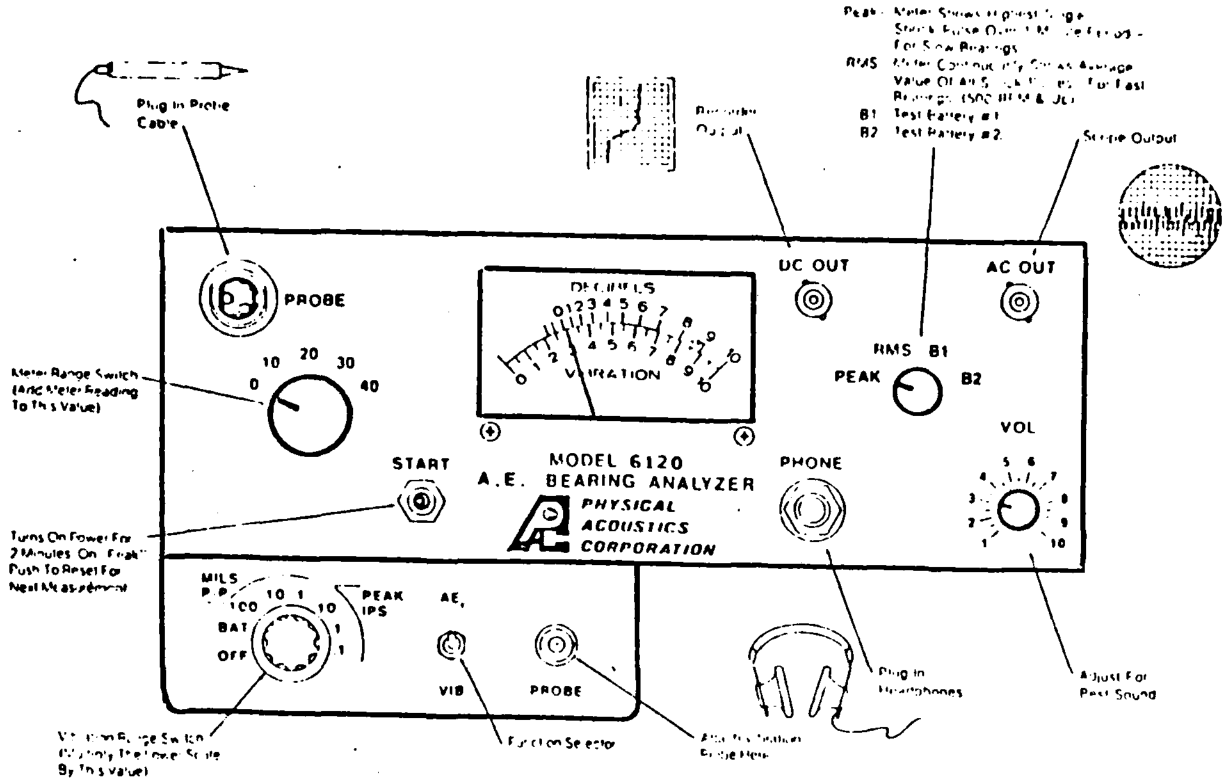


Figure 3.7: Controls of Model 6120 Bearing Analyzer [14]

Measuring Range:	0 to 50 dB
Meter:	0 to 10 dB - linearity ± 0.5 dB
Output-DC (full scale):	0.10 V DC into 100,000 ohms
Output-AC (full scale):	2.0 VRMS into 100,000 ohms
Power:	4-9V standard transistor battery

It should be noted that the Model 6120 is tuned to 40 KHz making it especially effective for bearings but less suitable for a wide range of applications. A tunable filter could be easily incorporated.

3.4 Acoustic Emission Transducers

3.4.1 Transducer Description and Selection

An acoustic emission transducer is a specialized piezoelectric sensor constructed to respond at high frequencies. The A.E. transducer is different from an accelerometer in several ways:

(1) The A.E. transducer may have several resonances and usually has a "spikey" frequency response.

(2) The accelerometer is usually designed with a specific resonance and damping factor so as to provide the most linear frequency response or, in some cases, shock response.

Accelerometers are often used as "A.E." transducers in machinery health monitoring applications. The linear broad-band response would be desired in some applications. For high-frequency testing (above 40 KHz), accelerometers are generally less sensitive than normal A.E. transducers.

Figure Nos. 3.8, 3.9, and 3.10 contain the spark impulse calibration plots from three A.E. transducers used during this project. As can be

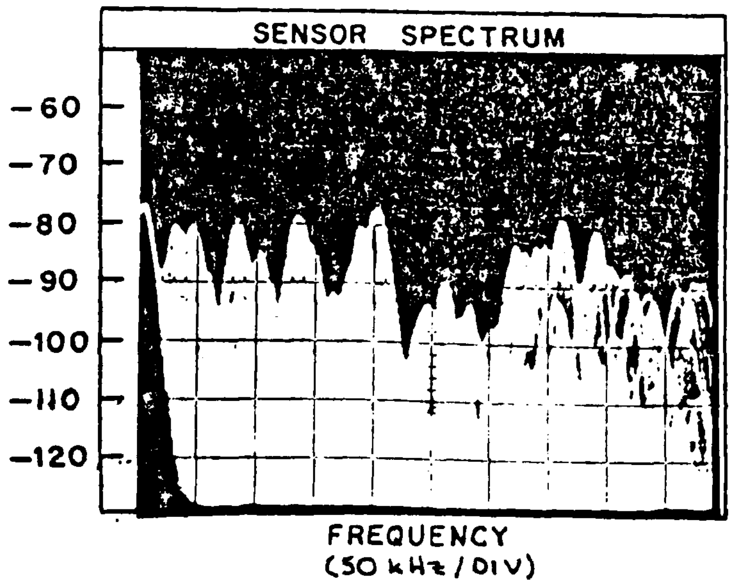


Figure 3.8: Calibration Record of PAC A3-350 Transducer

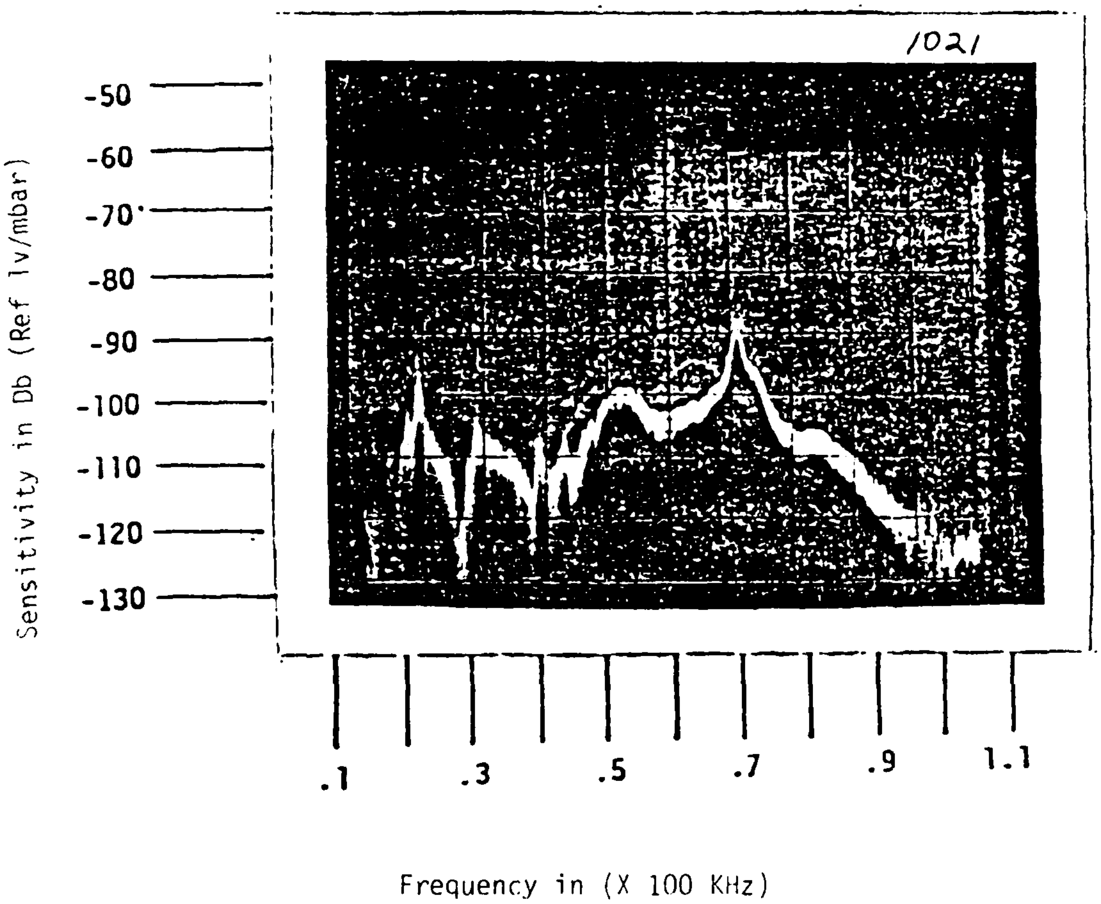


Figure 3.9: Calibration Record of AC 75L Transducer
Nominal Resonance 75 KHz

TRANSDUCER MODEL S9204 SERIAL NO. AA06

PEAK SENSITIVITY 2006.0 V/(m/s) or 66.0 dB re 1 V/(m/s)

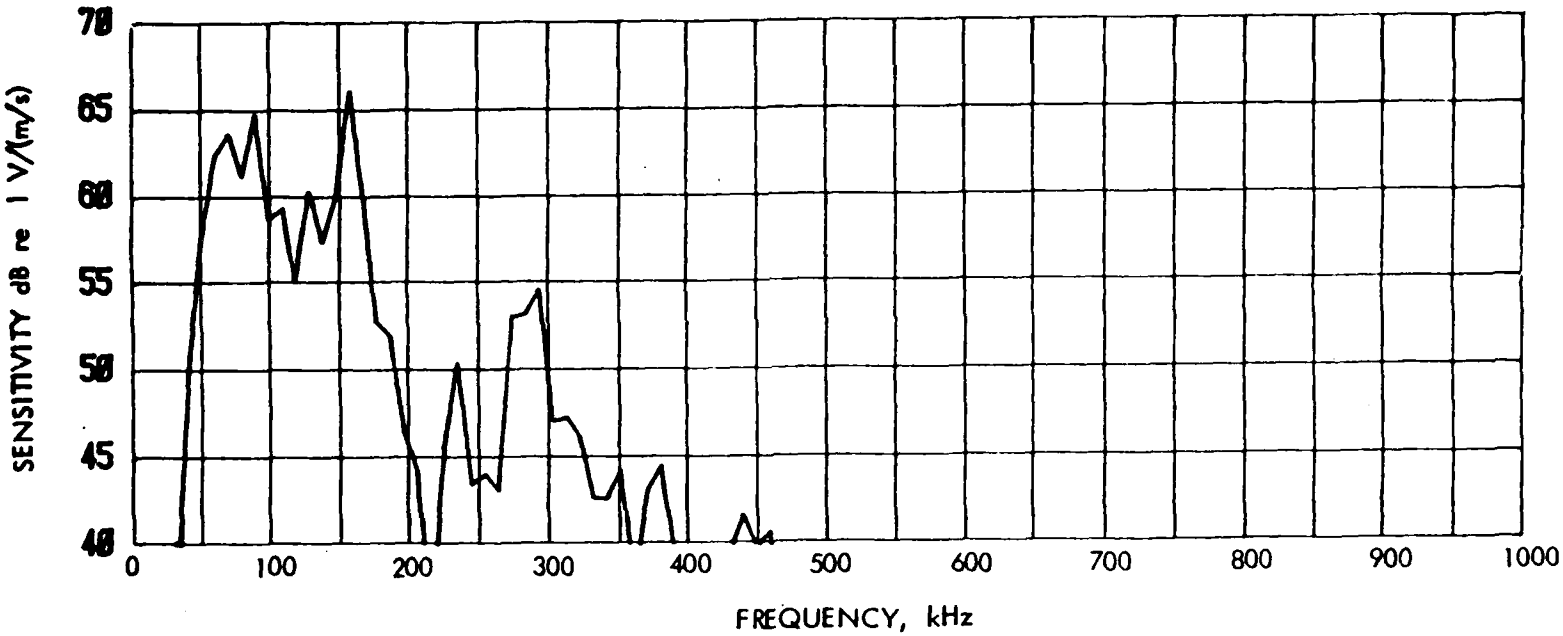


Figure 3.10: Calibration Record of Dunegan/Endevco S9204 AA06 Transducer

seen from these Figures, all transducers have a very nonlinear frequency response. The regions on the plots where the transducer exhibits a flatter response are considered the best region for obtaining acoustic emission information.

All A.E. transducers were chosen for each application by a trial-and-error procedure. This procedure basically involved testing each component with several different transducers and determining by visual inspection which transducer produced the "best" signal.

3.4.2 Transducer Mounting

Many products are offered by the "A.E." vendors for coupling transducers to the surface of a test component. Silicone fluids and greases are widely used for A.E. work in fatigue and nondestructive testing. These couplants, however, are often unsatisfactory for machinery diagnostic testing. Since the housings of pumps, valves and other test components are usually castings with rough surfaces, it is difficult to attain repeatable bonds with the transducer. One company offers brass transducer adaptors that can be cemented to a ground surface, providing a platform for a spring- loaded removable mount.

Many different couplants and mounting techniques were studied. The most consistent method was determined to be using a brass disk as an adaptor. The disk was filed to make good contact with the housing/ body of the test component. Cyanoacrylate cement was used to bond the disk to the transducer and then to the test component. This method provided repeatable results.

The addition of the adaptor block has an effect on the performance of the transducer, since its acoustic impedance is different. The mounted resonant frequency can easily be found by gently "thumping" the transducer with a pencil while observing the spectrum of an FFT analyzer. If the transducer-mount combination shows a sharp peak, then this frequency must be carefully examined in frequency domain or time domain records. While the spark impulse calibration plots are for the unmounted transducer (minus brass adaptor), these plots still proved valuable as an indicator of the range of frequencies where transducer resonances could be expected.

3.5 Spectrum Analyzer (SD 345 Spectrascope III) [15]

This research work made extensive use of a SD 345 Spectrascope III (spectrum analyzer) manufactured by Spectral Dynamics. This instrument has several operational modes. The mode most used for our work was the Fast Fourier Transform (FFT) signal processing mode. The following section gives a general description of FFT signal processing and how the analyzer works.

3.5.1 FFT Signal Processing

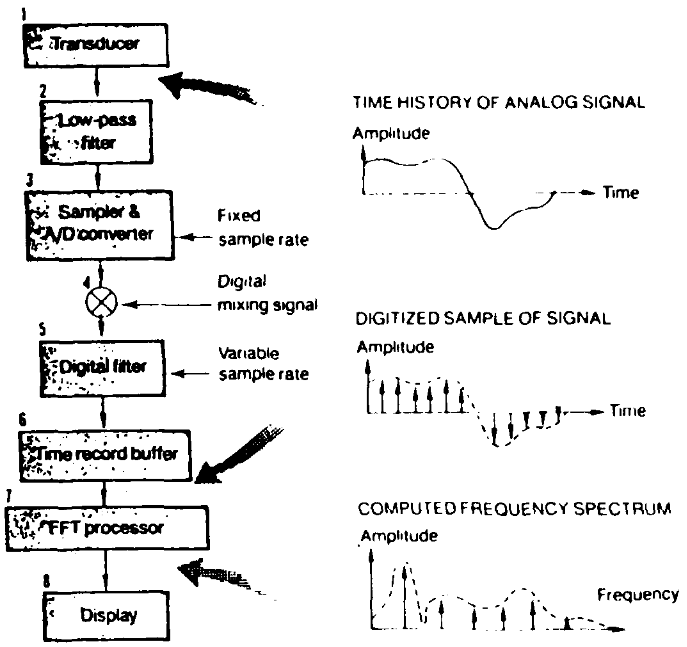
Real-time analyzers have a Fast Fourier Transform processor that implements a sophisticated computational algorithm for converting signal time histories into frequency spectra. This FFT processor cannot transform data in a continuous manner, so digitized samples of signals must be employed to form time records. These transformed samples in frequency spectra are called lines. The SD 345 has 400 lines for the computation and display of frequency spectra.

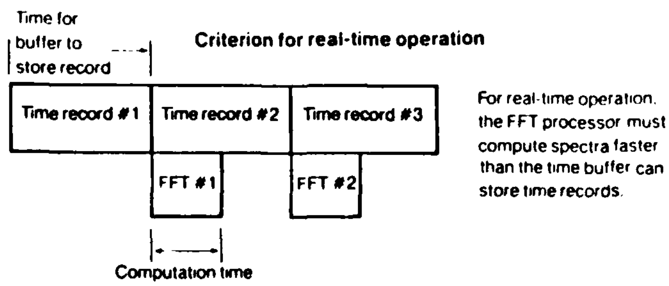
Sampled time records are processed as blocks of data. Consequently, a complete time record is needed for the FFT processor to compute each line in a spectrum. As a spectrum is computed, a new time record is sampled and stored in a buffer.

Finite time is needed to compute the FFT of a digitized time record. If the FFT is computed before the time buffer is filled, it can be assumed that no essential input data are missed. This means that measurements are made and spectra displayed in so-called real time.

The number of samples in a time record is fixed by the configuration of the FFT processor. However, the time record duration need not remain constant because the analysis span of the measurements can be adjusted. For measurements with a narrow frequency range providing high resolution, the sampling rate must be slowed to give long time records. On the other hand, the sampling rate must be increased to cover higher frequencies.

Figure 3.11 contains a block diagram description of a simplified single-channel FFT analyzer.





How the analyzer works

System component	Function
1	Converts vibratory motion into electrical signal
2	Filters spurious frequency components
3	Samples and digitizes analog signal
4	Facilitates zoom analysis
5	Filters spurious frequency components
6	Stores digitized time records
7	Computes frequency spectra
8	Displays spectra

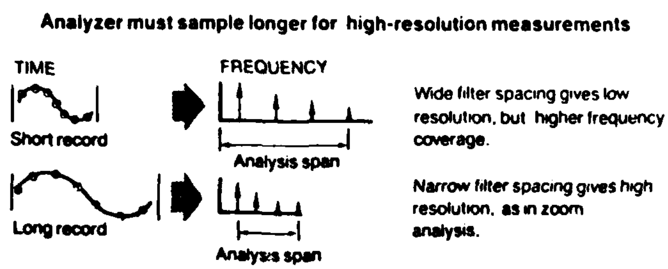


Figure 3.11: Block Diagram of Simplified Single-Channel FFT Analyzer [16]

CHAPTER IV

LABORATORY TESTS AND RESULTS

4.1 Hydraulic Cylinder Test [15]

Initial experiments attempting to measure the A.E. signal generated by leakage in a cylinder were unsuccessful due to the excessive noise generated by the supply pump. So that hydrostatic testing of the cylinders was possible. a large accumulator was used to supply oil to the cylinder. A sketch of the experimental apparatus used for the hydraulic cylinder test is shown in Figure 4.1. A test fixture was built in which the cylinder was mounted. The fixture held the cylinder in a fixed position when pressurized. Flexible hydraulic hoses were used to connect the cylinder to the system. These hoses reduced any structure-borne noise and also added to the ease with which the cylinder was connected or removed from the system. Hydraulic oil (MIL 2104) at 1500 psig and 100^{o}F was supplied to one side of the cylinder. The pressure was held constant by using a large accumulator. After the required supply pressure was attained, the pump was turned off, and the cylinder was supplied by the accumulator. This reduced the system fluid-borne noise. Oil from the nonpressurized side of the cylinder was collected and timed with a stop watch to measure the leak rate. To prevent any foaming of the oil in the return side, the line at that end was fitted with a riser in order to hold a head. The PAC A3-350 piezoelectric transducer was mounted on the surface of the cylinder

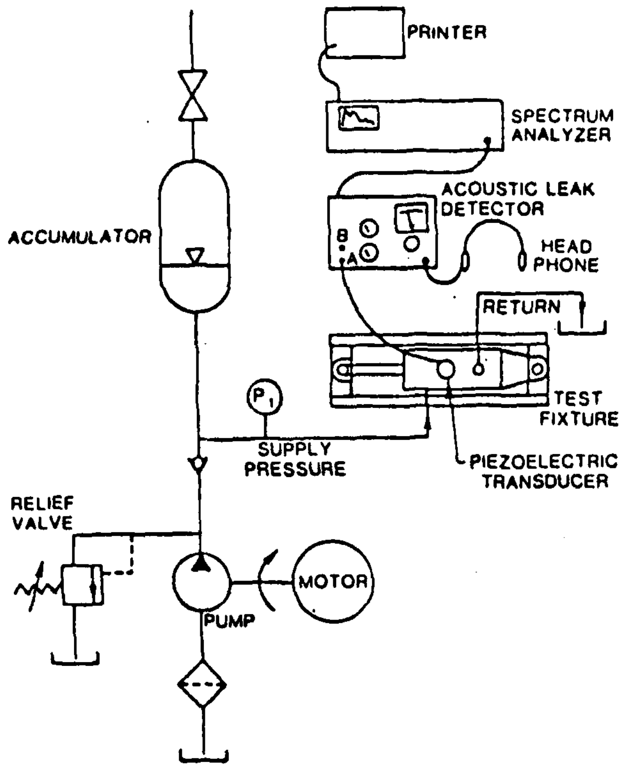


Figure 4.1: Schematic Diagram of the "Cylinder Leak Detection" System

directly over the seal. The transducer cemented to the cylinder provided a rigid bond between the transducer and cylinder housing. The signal from the transducer was amplified through the acoustic valve leak detector (5120) and was analyzed using a spectrum analyzer (Spectrum Dynamics - SD345 Spectrascope III). A hard copy of the frequency spectrum for 0 - 100 KHz of the amplified A.E. signal was obtained using a video printer (Axiom EX-850).

To simulate different leak rates, the seal of the piston was damaged externally and reassembled. All tests were performed with the same supply pressure and fluid temperature. Numerous graphs of acoustic amplitude versus frequency were obtained for leak rates ranging from 15 ml per min to 500 ml per min.

Two different but similarly sized cylinders were tested to check for repeatability. Though the exact same leak rates were not attained for both cylinders, the same range of leak rates was attained.

Spectral analysis was used as the signal analysis technique for all cylinder tests performed. This signal processing technique was chosen because of the positive results obtained by Dickey [10] using this technique in the valve leakage tests he performed.

4.1.1 Data Analysis of Hydraulic Cylinder Results

Figure 4.2 shows a typical frequency spectrum obtained using the spectrum analyzer. The bottom spectrum in the figure is the reference signal corresponding to a zero leak rate but maintaining the 1500 psig supply pressure on one side of the cylinder. The top spectrum in Figure 4.2

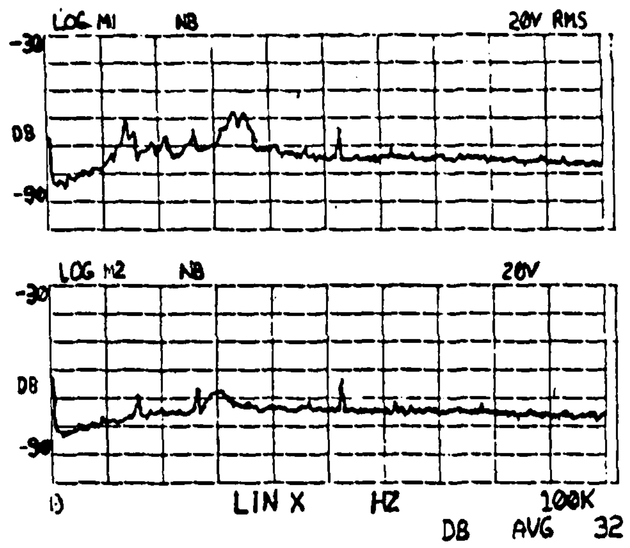


Figure 4.2: Typical Spectrum Analyzer Output for Cylinder Test

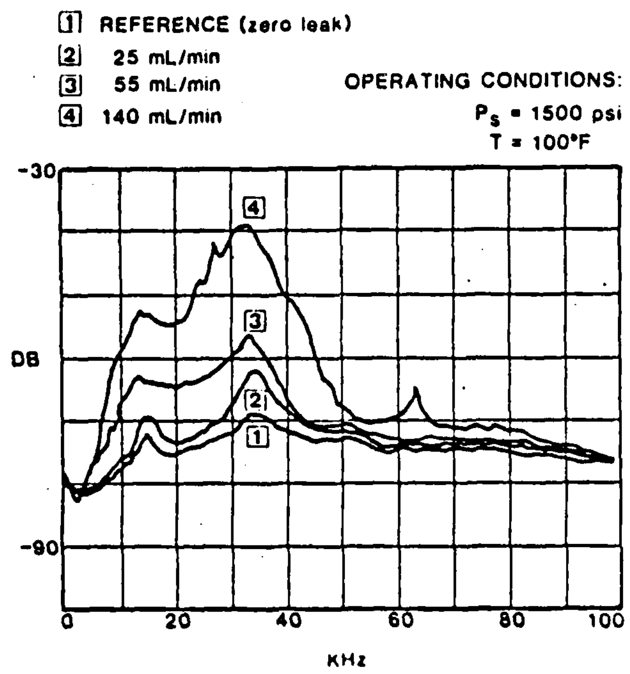


Figure 4.3: Frequency Spectra for Four Different Cylinder Leakage Rates Across the Seal

corresponds to a leak rate of 25 ml per min. Around 30 to 40 KHz, there is a peak corresponding to the nominal resonance frequency of the transducer. The amplitude of this peak can be taken as an indication of the leak rate. Figure 4.3 contains the frequency spectrum for different simulated leak rates.

The RMS value of the A.E. signal from 0 - 100 KHz was plotted versus leak rate for the two cylinders. The result is shown in Figure 4.4. The plotting is done on a logarithmic scale for leak rate on the Y axis. The A.E. signal output is presented as a RMS value in decibels with the reference level as 1 volt per meter per second squared. From Figure 4.4, it can be seen that the RMS value increases with increasing leak rate.

The experimental results show that the RMS value of the A.E. signal for the frequency range of 0 to 100 KHz is indicative of the leak rate. The information derived from the frequency spectrum of the A.E. signal can be used as a criterion for the acceptability or rejection of a cylinder.

It should be remembered that, in a leak detection system, the transmission of the sound between the source and the detector may play an important role in the determination of the actual relation between the A.E. output and the leak rate. The vibration caused by leakage will decay away from the source. It is therefore necessary to take all recordings with the transducer in the same position. Variation of the position of the transducer would cause a corresponding variation in the amplitude of the A.E. signal, which would not reflect the true

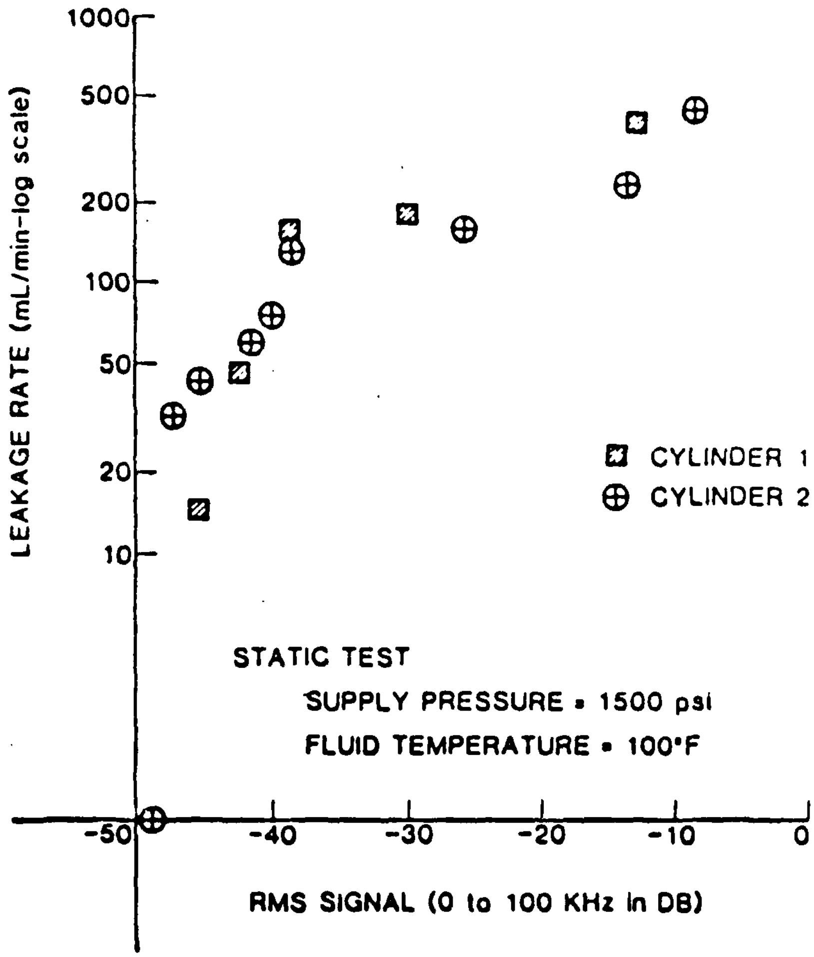


Figure 4.4: Experimental Results of Cylinder Tests

nature of the leak-related noise. The position used for transducer mounting for all cylinder tests performed was on the cylinder housing directly over the seal. This position proved the best for measuring the leak-related noise.

4.2 Valve Test

As in the cylinder tests, preliminary attempts to measure changes in valve leakage rates were unsuccessful because of the masking by the fluid-borne pump noise by the power supply. Since the desired leak rates were so low (10 to 50 ml/min), an accumulator was incorporated, making possible a hydrostatic test of in some instances several minutes duration. The system is similar to that used in the cylinder leakage tests. A sketch of the experimental apparatus used for the valve tests is shown in Fig. 4.5.

The valve tested was a "Dayton" four-way hydraulic, directional-control valve having a single spool and manual actuation. A cross section of the valve is shown in Fig. 4.6.

The valve was only tested for leakage across the spool. To produce leakage of this type, the control ports of the valve were blocked so that flow could only occur across the spool. The inlet of the valve was supplied with hydraulic oil (MIL 2104) at 1500 psig, and the outlet of the valve was kept at atmospheric pressure.

The test valve in its initial condition had no measurable leakage across the spool when the spool was in the neutral position. Two methods were used

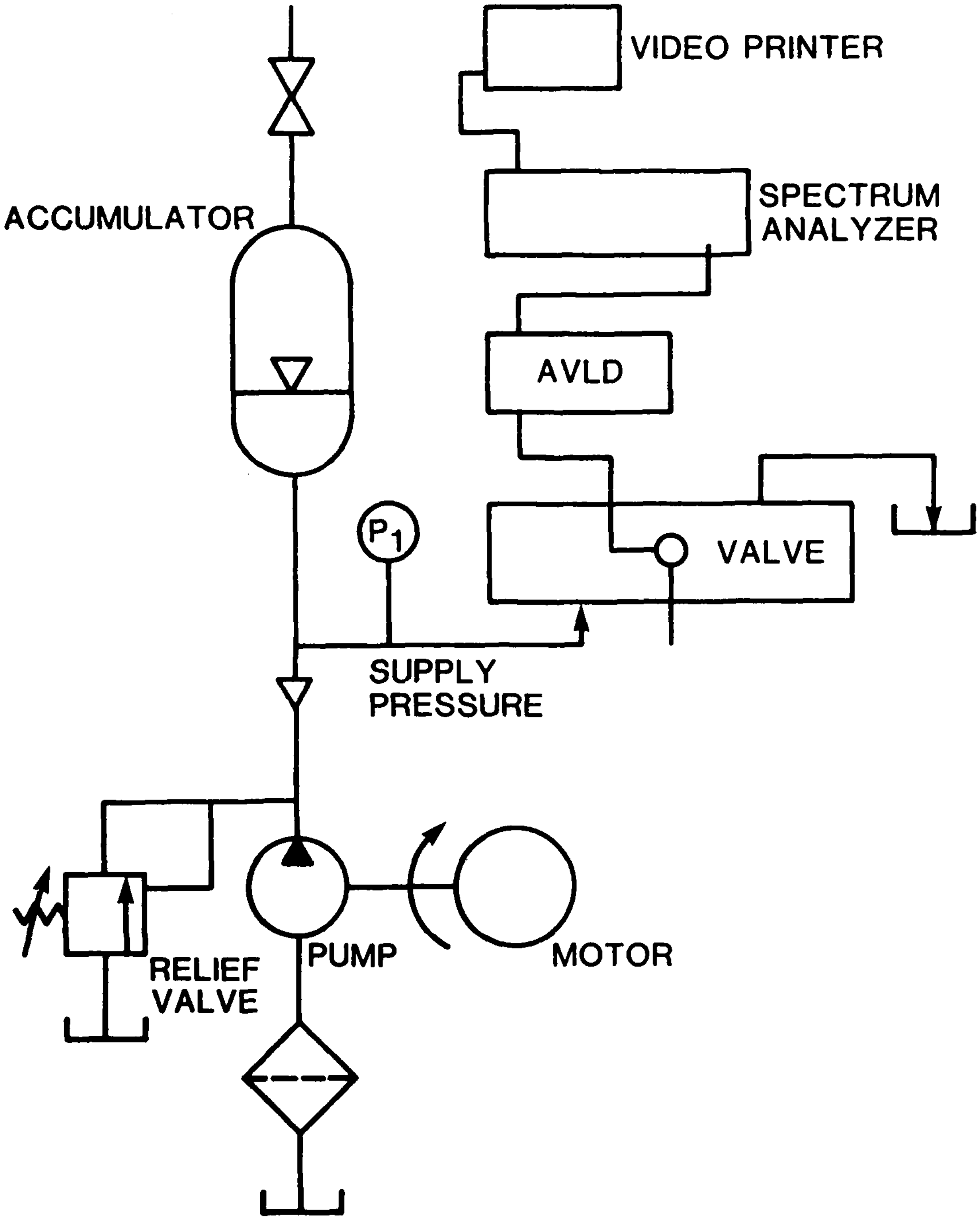


Figure 4.5: Schematic Diagram of the "Valve Leak Detection" System

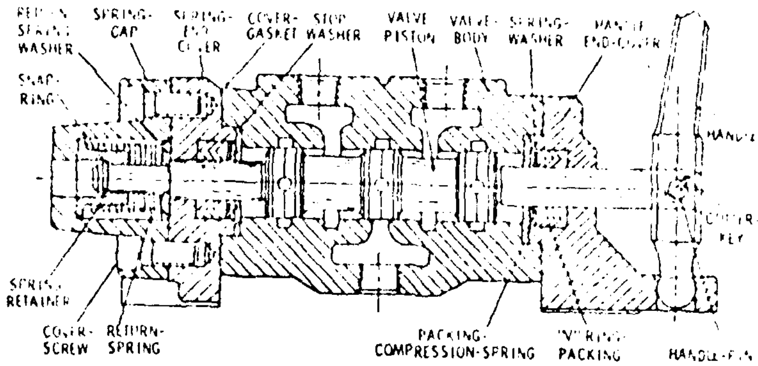


Figure 4.6: Valve Schematic:

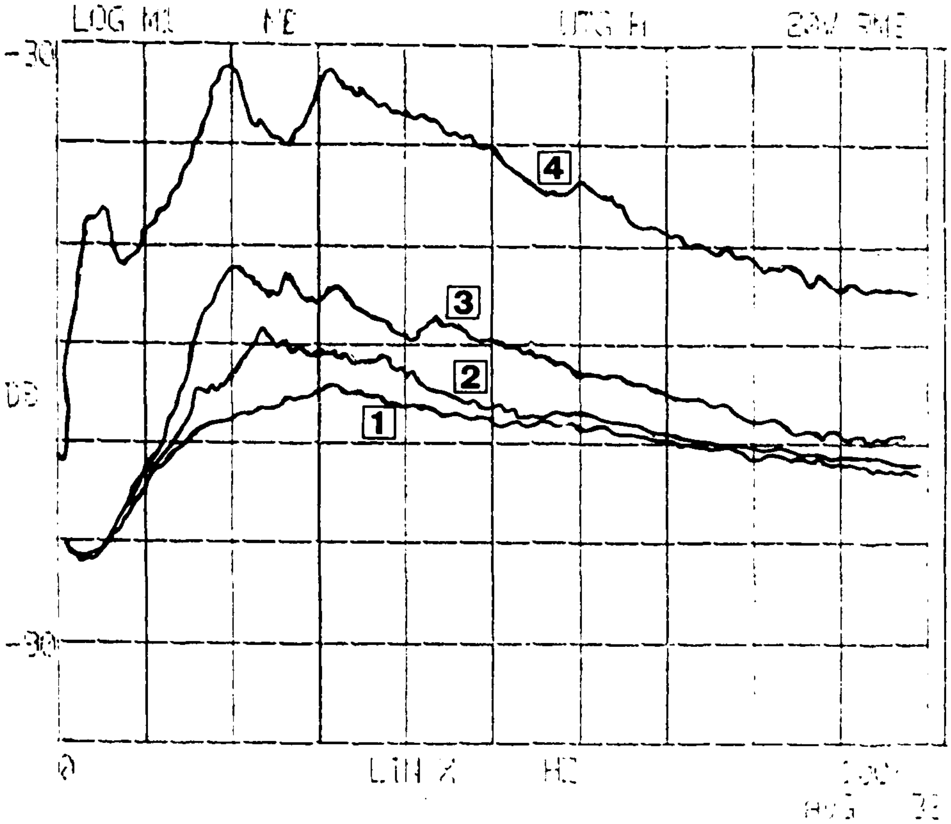
Single Spool Spring Centered Manually Operated Directional Control Valve to create leakage across the spool. The first method involved mechanically displacing the spool from its neutral position. Different leak rates were obtained by displacing the spool by various amounts. The second method involved degrading the valve on a separate test stand using standard contaminants. Degradation of the valve caused leakage to occur across the land of the spool when the spool was in the neutral position. Various leak rates were obtained by testing the valve at different degrees of degradation.

The accumulator of the experimental test stand was charged by a pump and when the desired pressure was obtained, the pump was turned off and the valve was supplied by the accumulator. This reduced the system generated noise. Oil from the nonpressurized outlet of the valve was collected and timed with a stop watch.

The PAC A3-350 piezoelectric transducer used in the cylinder test was also used for these tests. The transducer was cemented to the housing of the valve between the blocked control ports. The signal from the transducer was amplified through the acoustic valve leak detector and analyzed using a spectrum analyzer.

4.2.1 Valve Test Results

Figure 4.7 contains the narrow-band spectra of the A.E. signals produced by leakage in the valve. This leakage was created by the displacement of the spool from its neutral position. The bottom spectrum in Fig. 4.3 corresponds to a minimum leak rate but maintains the 1500 psig



Frequency Spectrum For "VALVE LEAKAGE" Across Spool

(1) REFERENCE (Zero Leak)
(2) 16 ML/MIN
(3) 35 ML/MIN
(4) 170 ML/MIN

Figure 4.7: Frequency Spectrum for Valve Leakage Across Spool caused by Mecahnical Displacement of Spool (0-100 KHz)

supply pressure on the inlet side of the valve. The [illegible] figure corresponds to a leak rate of 170 ml per min. As can be seen from this Figure, the amplitude of the A.E. signal across the entire spectrum increases with increasing leak rate. There are two regions in these spectra which exhibited the greatest sensitivity to the leak-related noise. One region is a narrow band of frequencies centered around 20 KHz. The second region is the peak between 30 and 40 KHz, which corresponds to the natural frequency of the transducer when mounted on the cylinder. Both of these regions could be used as an indicator of the leak rate in a valve for this type of leakage.

Figure 4.8a-b contains the spectra obtained from the valve leakage tests involving the valve degradation. Figure 4.8a corresponds to a leak rate of 39 ml per min. These figures reveal that this type of leakage-produced noise is sensed mainly in a narrow band of frequencies centered around 35 KHz. Comparison of these spectra to the spectra shown in Fig. 4.7 also reveals that this leakage noise is of a smaller magnitude than the leakage noise produced by spool displacement.

The difference between the spectra obtained from the two leakage tests can be explained by considering the types of flow involved. In the leakage tests using spool displacement, the leakage flow produced would be similar to flow through a sharp-edged orifice. This type of flow would be very turbulent in nature. For the leakage tests in which valve degradation was used, the leakage flow would be similar to flow through an orifice with finite width. The characteristic length of the orifice would be equal to the amount of overlap in the valve. Flow

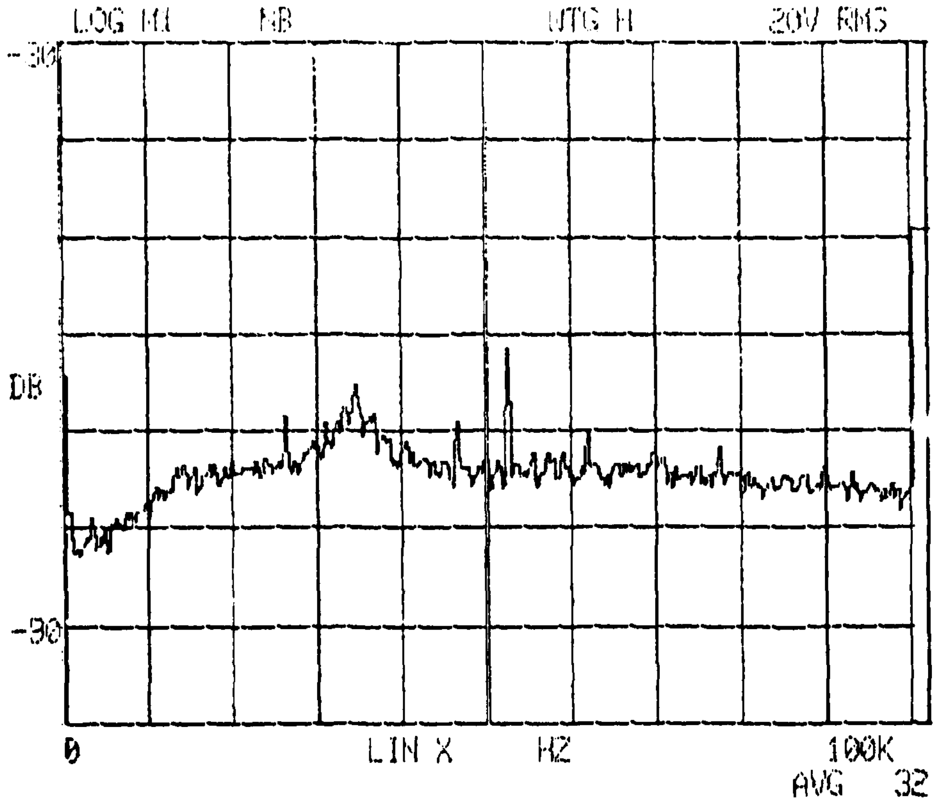


Figure 4.8 a: Frequency Spectrum for Valve Leakage Across Spool Caused by Degradation of Valve (0-100 KHz) - Zero Leak

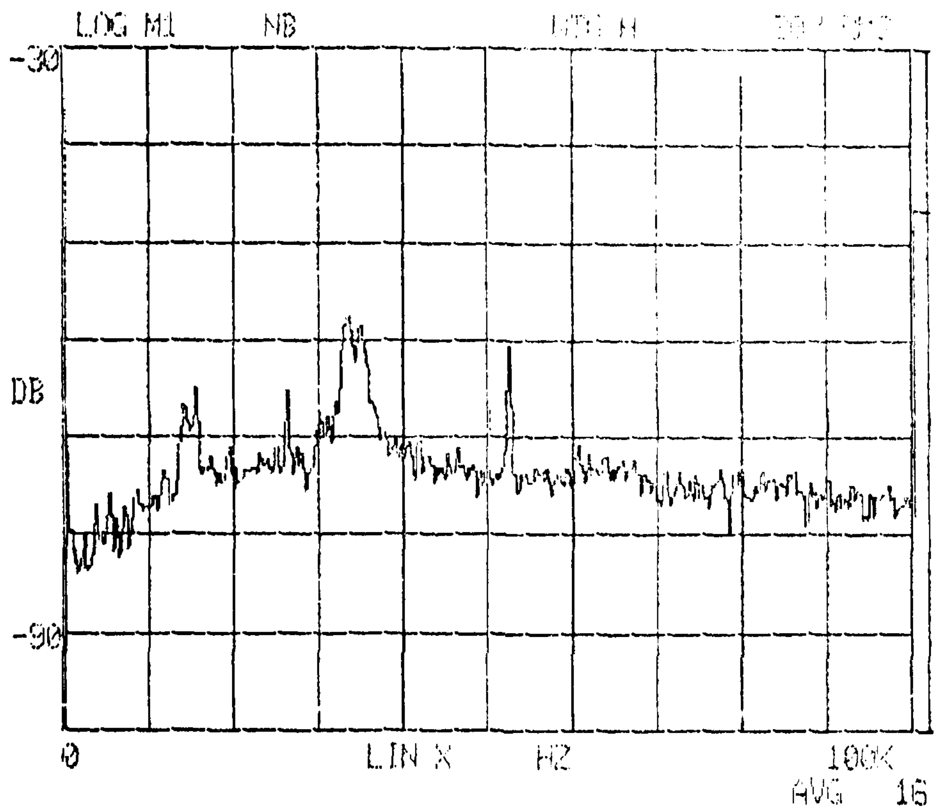
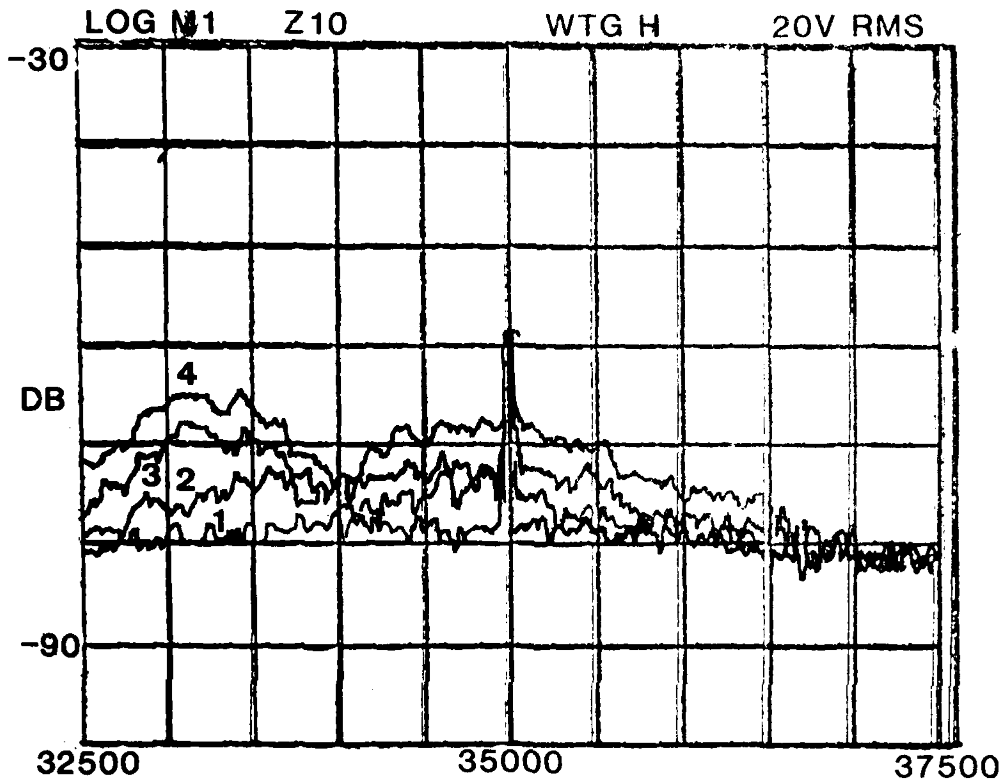


Figure 4.8 b: Frequency Spectrum for Valve Leakage Across Spool caused by Degradation of Valve (0-100 KHz) - 39 mℓ/mim leak

through this type of orifice would generally be less turbulent than flow through a sharp-edged orifice. It is presumed that the amount of turbulence present in the flow has a direct effect on the A.E. signal produced. In other words, a increase in turbulence in flow will cause a corresponding increase in the A.E. signal produced. This would then explain the difference in the results obtained from the two types of leakage tests.

Figure 4.9 contains the narrow-band spectra from 32.5 to 37.5 KHz of the A.E. signals obtained during the degradation tests. These spectra show that, by zooming in on the 35 KHz peak mentioned previously, the leakage noise for leak rates as small as 6 ml/min can be measured. The bottom spectrum of Fig. 4.9 corresponds to a zero leak rate. The top spectrum corresponds to a leak rate of 39 ml/min. This figure shows that the average amplitude between approximately 32.5 KHz and 35 KHz would be a good indicator of the leakage present in the valve.

The results of these tests show that there are measurable A.E. signals related to the leakage present in a valve. The mechanism used to create leakage was determined to affect the resulting A.E. signal. The leakage mechanism had an effect on the turbulence present in the flow, and this was concluded to have a direct effect on the A.E. signal produced. Spectral analysis of the A.E. signals for both sets of tests yielded information indicative of the leakage present.



(1) - ZERO
(2) - 6 ML/MIN
(3) - 25 ML/MIN
(4) - 39 ML/MIN

Figure 4.9: Zoom Frequency Spectra for Different Valve Leakage Rates caused by Degradation of Valve (32.5 to 37.5 KHz)

CHAPTER V

HYDRAULIC PUMP TESTS

This section is concerned with the application of acoustic emission techniques to the diagnostic testing of hydraulic pumps. Experiments were performed on vane pumps, piston pumps, and gear pumps. A.E. tests for all pumps were performed on a pump test stand. Two gear pumps were also tested on a field unit. The test stand experiments were designed to determine the types of pump defects that could be detected from an A.E. signal. The field tests were designed to verify that pump defects which proved to be detectable on the test stand could also be detected on an actual field unit. The field unit used for these experiments was a 2 1/2 cu. yd. scoop-type loader.

The pumps were tested for numerous types of defects or process problems. These are:

(1) Pump degradation by contaminants

(2) Pump cavitation

(3) Pump internal mechanical damage (shaft damage, bearing damage, gear tooth damage, etc.)

Before any damage or degradation was inflicted on the pumps, the baseline A.E. signatures were obtained for later comparison purposes. In addition, the volumetric efficiency curves were obtained for the test pumps to establish the initial leakage condition. The volumetric efficiency curves of the degraded pumps were also obtained so that the amount of degradation inflicted upon the pumps could be specified quantitatively.

The following sections contain the specific experimental procedures of each test and the results obtained.

5.1 Vane Pumps

5.1.1 Degradation Tests

Two identical vane pumps were tested on the pump test stand for this series of experiments. The specifications for these pumps are given in Appendix D. One pump, pump B, was obtained in a degraded condition. The other pump, pump A, was obtained in its new condition.

The A.E. transducer used during these experiments was the AC-75L. The transducer was cemented to the housing of the pump, and its output was amplified using the AVLD. The amplified signal was analyzed using spectral analysis with a SD 345 Spectrascope III spectrum analyzer. The narrow-band spectrum from 0 to 100 KHz of the A.E. signal was recorded at several different pressure conditions. The pump shaft speed and the hydraulic fluid temperature were held constant for these tests.

A comparison of the volumetric efficiency for both pumps is shown in Fig. 5.1. It can be seen from this Figure that, at 1500 psig, there is a 20 percent difference in efficiency between the two pumps. Because of this difference, it was expected that there should also be a corresponding difference in the A.E. signal produced by the two pumps.

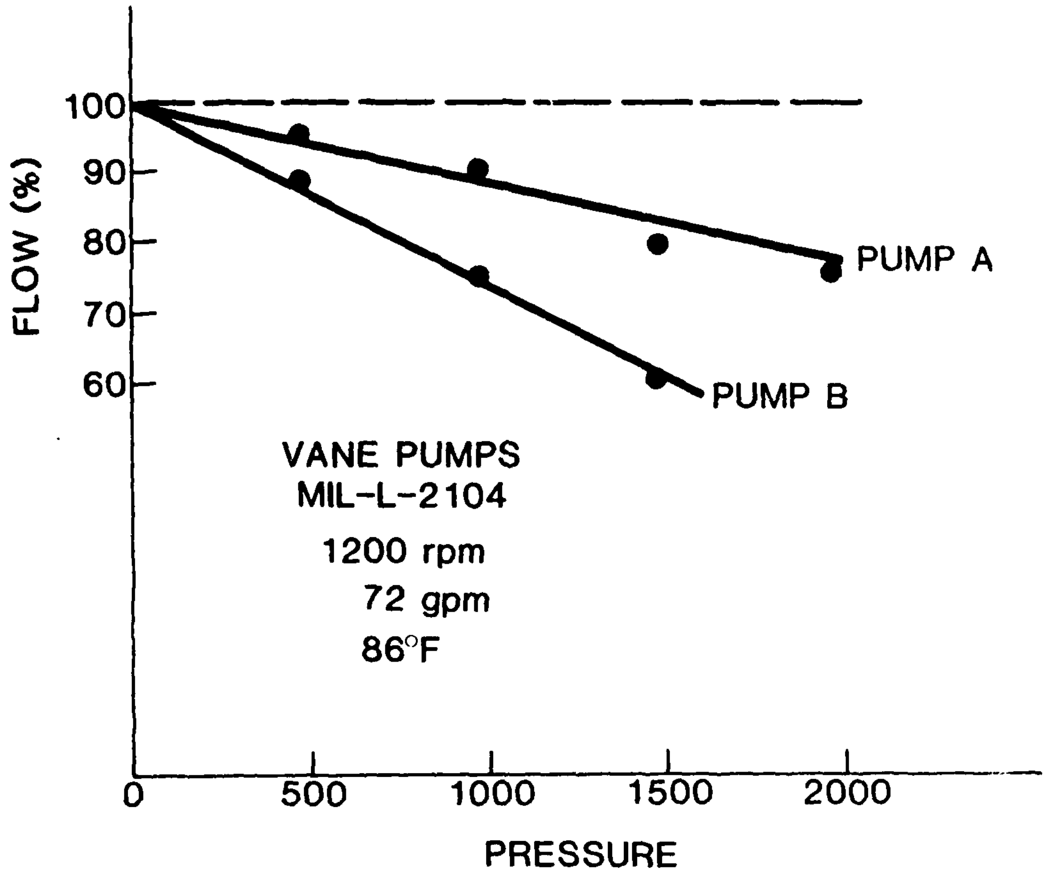


Figure 5.1: Pump Volumetric Efficiencies - Vane Pumps

The spectra of the A.E. signal obtained from the two pumps for two outlet pressures, 0 and 1500 psig, are shown in Figs. 5.2 and 5.3.

Figure 5.2 contains the spectra obtained for the zero outlet pressure condition. For this outlet pressure, both pumps produced the same output flow. As can be seen from this figure, the RMS values of the A.E. signals for both pumps are approximately the same. Comparison of these spectra reveals that there are only minor differences between the two spectra. At 70 KHz, a peak occurs in the spectra of both pumps. This particular frequency is the nominal natural frequency of the transducer when mounted on the pump. A 6 dB difference between the good and the bad pump at this frequency was noted. This difference could be an indication of the difference in mechanical health of the pumps. However, the magnitude of the difference is too small when considering the fact that the bad pump was in a 20 percent degraded condition compared to the good pump.

Figure 5.3 contains the spectra obtained for the 1500 psig outlet condition. The output flow for the good pump was reduced by 20 percent at this outlet pressure, while the bad pump's outlet flow was reduced by 40 percent. From this figure, it is seen that the RMS value for Pump B, the degraded pump, has decreased slightly. The RMS value for Pump A, on the other hand, has remained approximately the same. The 70 KHz peak for the good pump has increased approximately 5 dB for this outlet pressure. The 70 KHz peak for the bad pump has decreased approximately 6 dB.

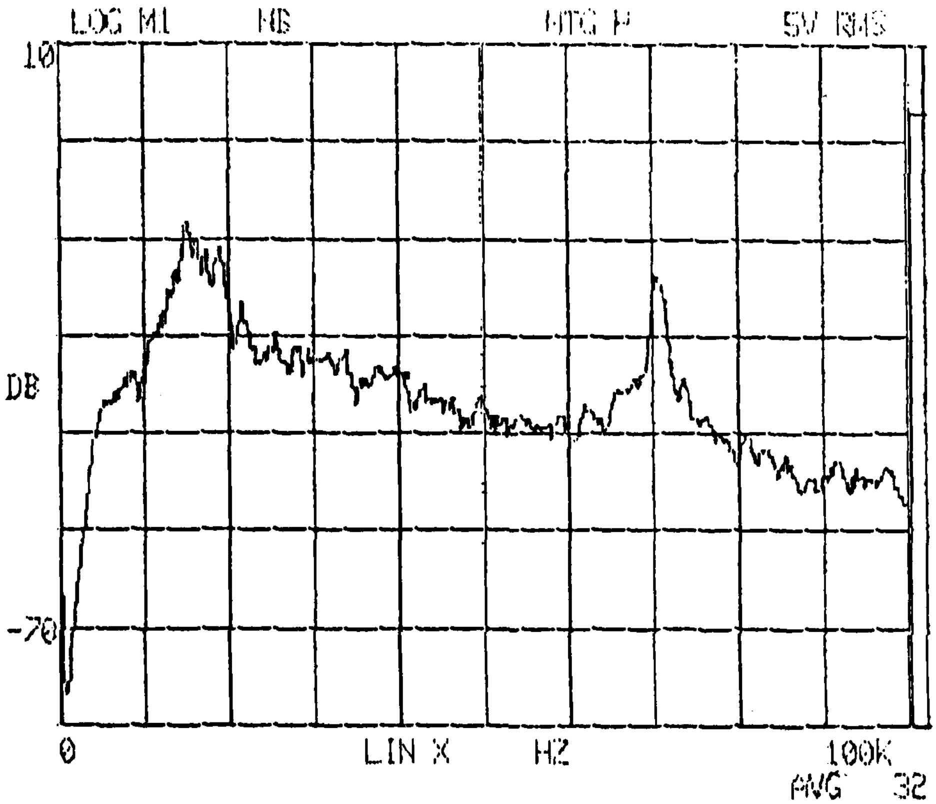


Figure 5.2 a: Frequency Spectrum Obtained from Vane Pumps Outlet Pressure 0 psig (0-100 KHz) Pump B

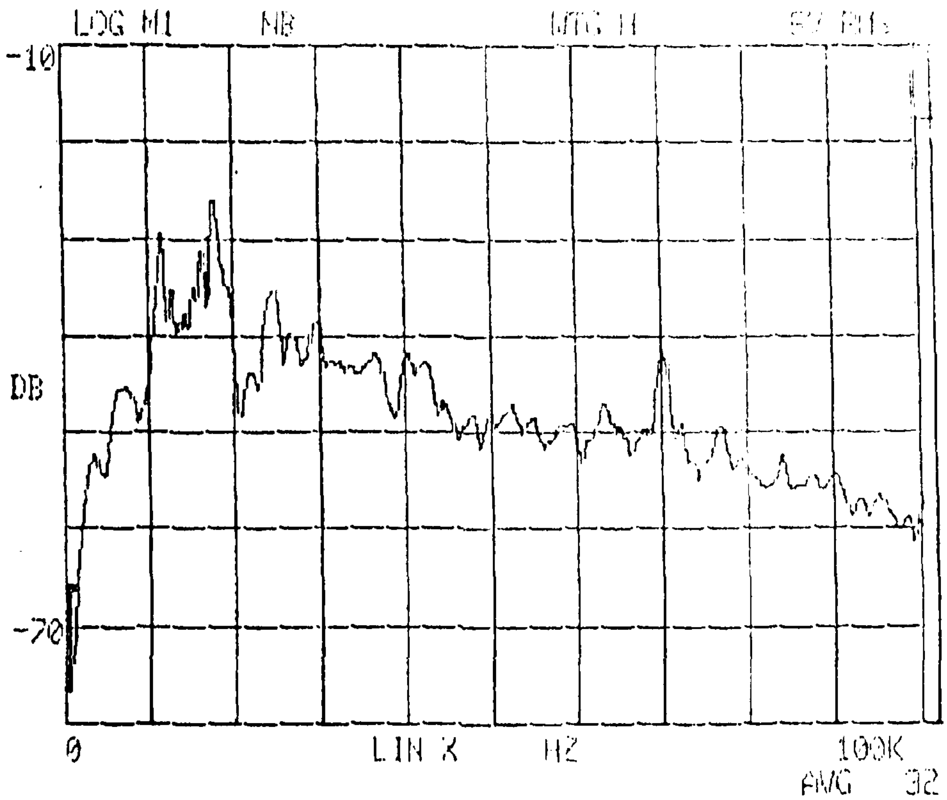


Figure 5.2 b: Frequency Spectra Obtained from Vane Pumps Outlet Pressure 0 psig (0-100 KHz) Pump A

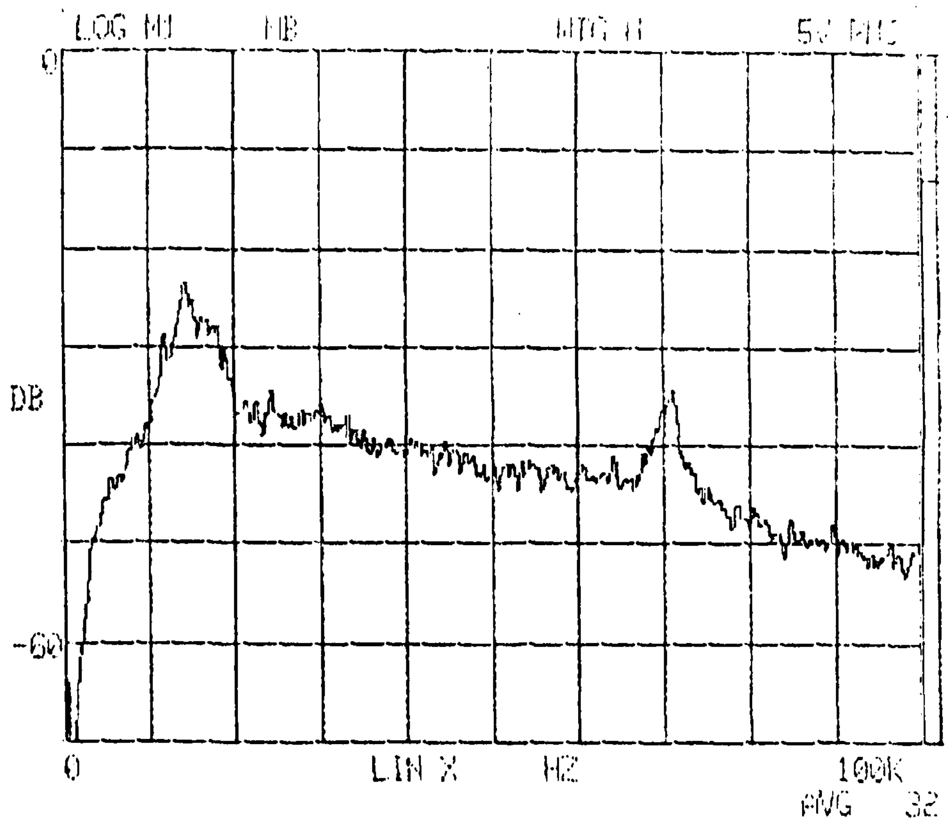


Figure 5.3 a: Frequency Spectra Obtained from Vane Pumps - Outlet Pressure 1500 psig (0-100 KHz) Pump B

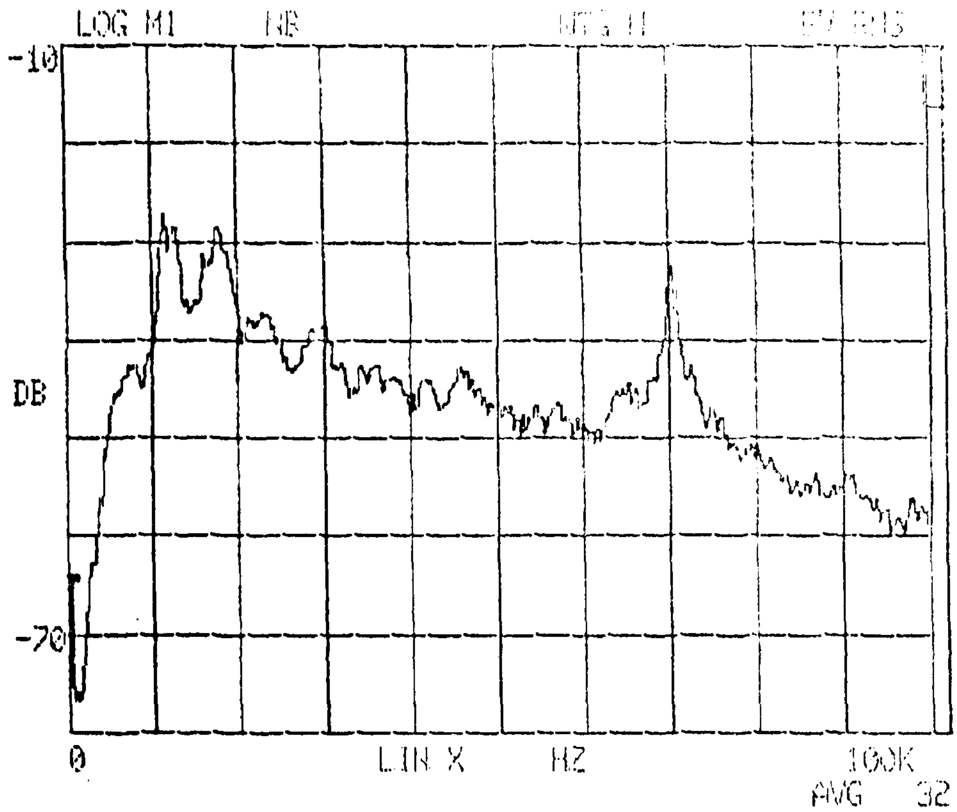


Figure 5.3 b: Frequency Spectra Obtained from Vane Pumps - Outlet Pressure 1500 psig (0-100 KHz) Pump B

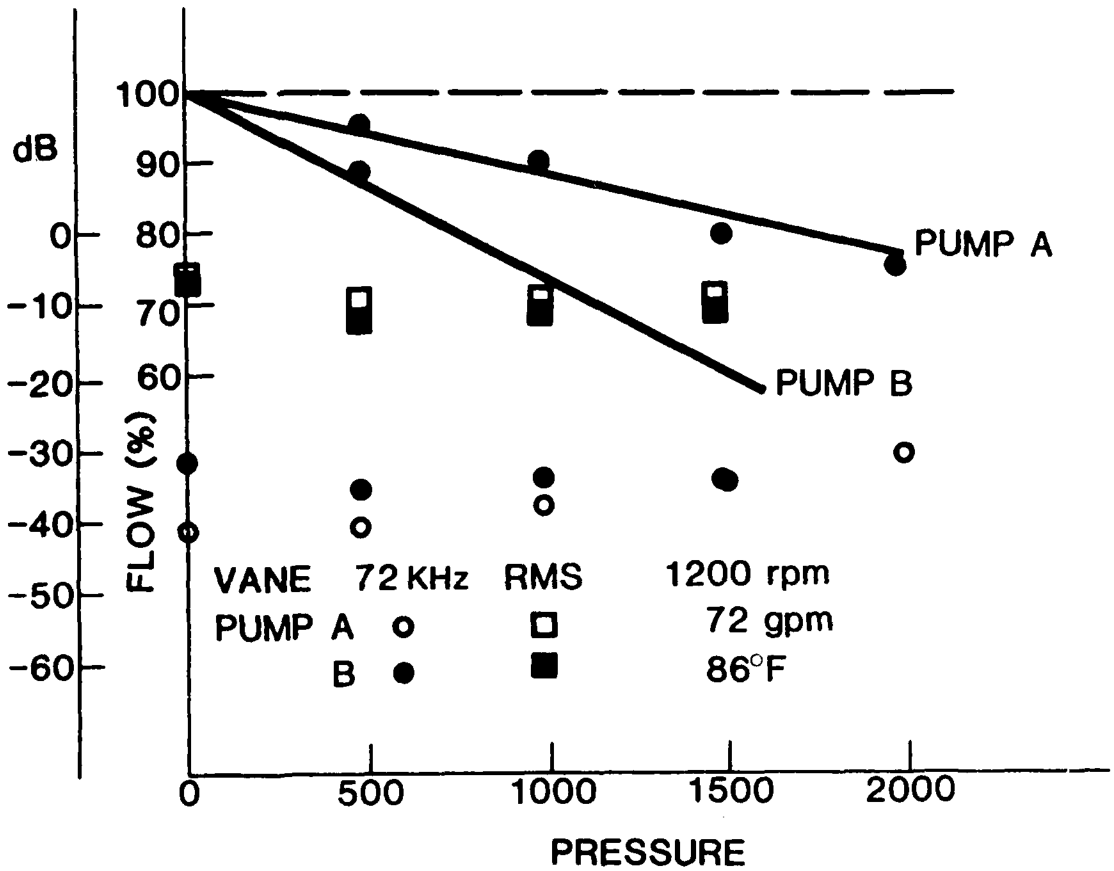


Figure 5.4: A.E. Signal Level VS. Pump Outlet Pressure - Vane Pumps (A & B)

Figure 5.4 contains plots of the RMS value versus outlet pressure and also the amplitude of the 70 KHz peak versus the outlet pressure obtained from the two pumps. The RMS value and the amplitude of the 70 KHz peak for the degraded pumps shows a slight decrease with increasing outlet pressure. These variables for the good pump showed a slight increase with increasing pressure. All of these variations, however, were 6 dB or less and were not considered significant.

Summarizing the results of this work, the following were determined.

(1) Degradation effects in pumps are not easily detectable from the A.E. signal.

(2) Degraded pumps exhibit a slight decrease in RMS level as well as the amplitude at 70 KHz with increasing outlet pressure.

(3) New/Good pumps exhibit a slight increase in the RMS level and in the amplitude at 70 KHz with increasing outlet pressure under specific test conditions.

5.1.2 Internal Mechanical Damage

The possible detection of a bearing or a bad vane in a vane pump with A.E. techniques was investigated in this se[illegible]es of tests. The first set of tests involved externally damaging a bearing of the vane pump with contaminants (AC Test Dust). Pump A was used for this test. Once reassembled with the bad bearing, the pump was operated at 0 psig outlet pressure and 1200 RPM. The AC-75L transducer was used for this experiment, and its output was amplified using the AVID. The narrow

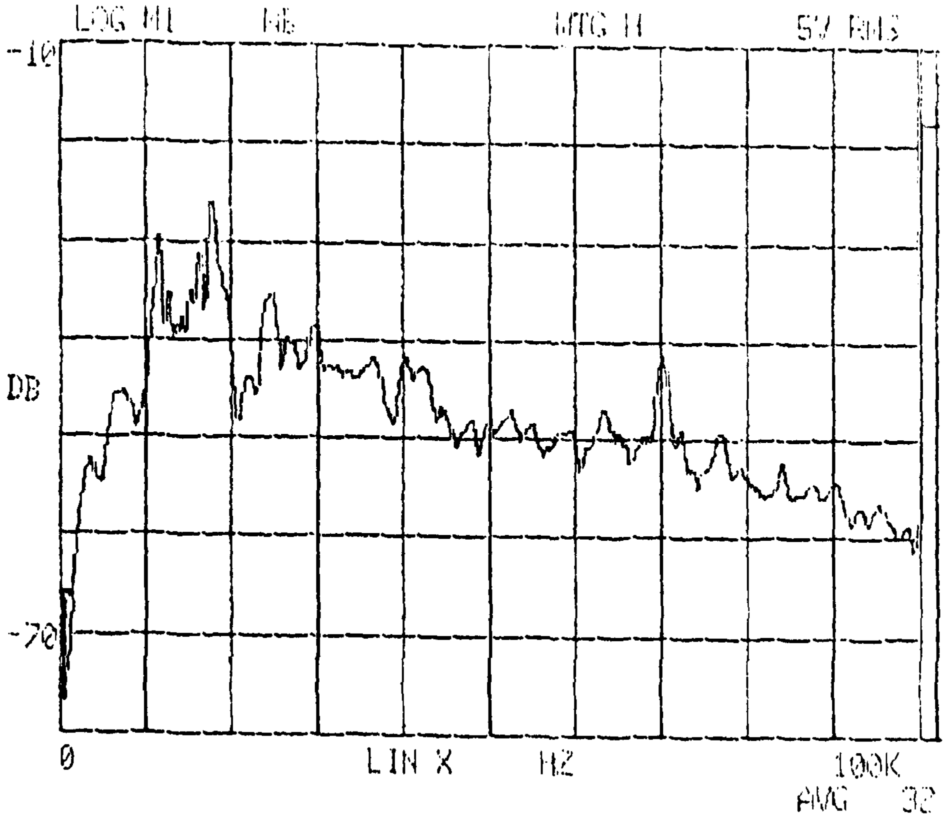


Figure 5.5 a: Frequency Spectrum of Vane Pump with Good Bearing

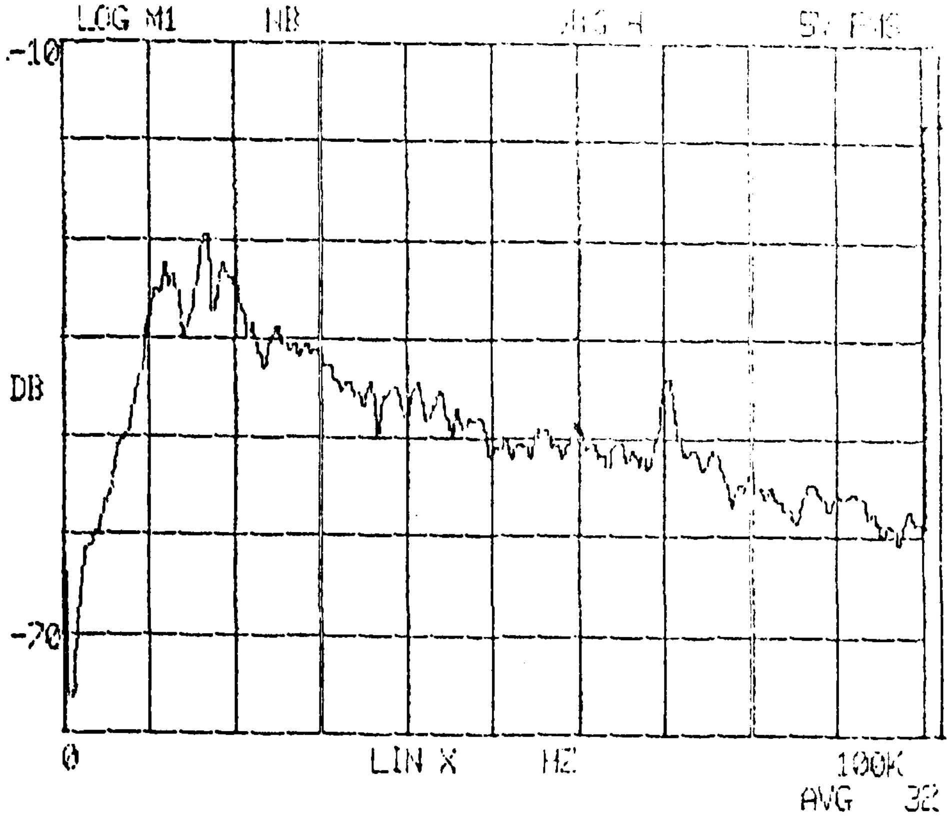


Figure 5.5 b: Frequency Spectrum of Vane Pump with Bad Bearing

band spectrum from 0 - 100 KHz of the A.E. signal was obtained using a spectrum analyzer.

Figure 5.5 contains spectra obtained from the pump in undamaged and damaged conditions. Only slight differences were observed between the two spectra. These differences were considered insignificant considering the amount of damage.

The second test involved damaging a vane in the pump. Again, the pump was operated at 0 psig outlet pressure and 1200 RPM. The A.E. signal was obtained in the same manner as before. The resulting spectra, again, did not yield any conclusive results.

5.2 Piston Pumps

Incipient Cavitation

Piston pumps are very sensitive to inlet pressure conditions, causing cavitation to be a major problem. Incipient cavitation detection would be useful to detect inlet pressure problems and prevent pump damage.

Incipient cavitation (and cavitation in general) produces high frequency vibrations and is well suited to detection using A.E. techniques. This set of tests involves the detection of incipient cavitation from the frequency spectrum of an A.E. signal.

A standard OSU-P-2 test was conducted for an axial piston pump. The specifications for this pump are shown in Appendix B. The AC-75L transducer was cemented to the housing of the pump, and its output was amplified using the AVLD. The narrow-band spectrum of the A.E. signal from 0 - 100 KHz was obtained using a spectrum analyzer. The fluid temperature and outlet pressure were held constant during these tests. Cavitation was produced by decreasing the inlet pressure to the pump.

Figure 5.6 contains two spectra obtained during these tests. The spectrum in 5.6 a was obtained from the pump under normal operating conditions (inlet pressure = 1.65 ATM). The spectrum in Fig. 5.6 b was obtained from the pump at the incipient cavitation point. As can be seen from this spectrum, the amplitude of the 70 KHz peak decreases when incipient cavitation occurs. This peak proved to be a very good indicator of incipient cavitation. Its amplitude indicated cavitation conditions long before standard flow measurements yielded cavitation information.

Figure 5.7 shows the variation of the amplitude of the 70 KHz peak decreases to a minimum at the point of incipient cavitation. Further reduction of the inlet pressure at this point causes an increase in this amplitude. The test was terminated when cavitation became obvious.

It was also observed that the amplitude variation at 9 KHz is also shown in Fig. 5.7. This amplitude shows the same temdency as the 70 KHz peak.

In conclusion, the use of the A.E. signal in detecting incipient cavitation in pumps proved to be a very sensitive technique. The amplitude of the A.E. signal at 70 KHz could be used to establish minimum inlet pressure requirements for non-cavitation conditions. Several experiments were also conducted relating to cavitation in gear pumps.

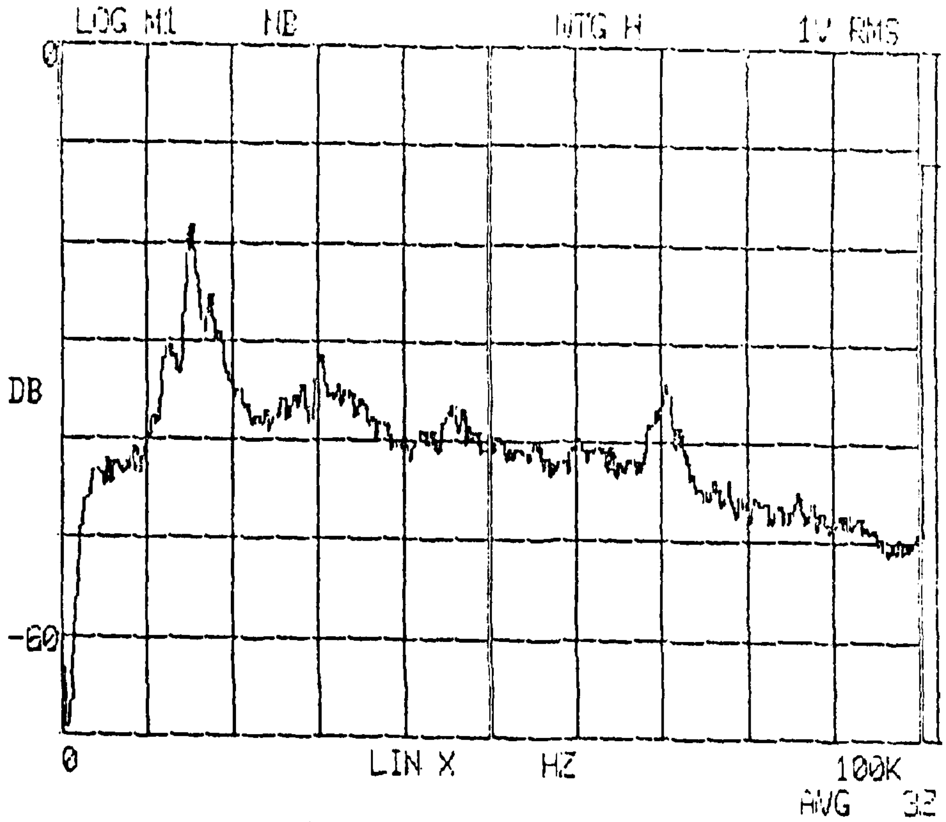


Figure 5.6 a: Frequency Spectrum (0-100 KHz) of Piston Pump. Cavitation Test. Inlet Pressure = 1.55 atm (Normal)

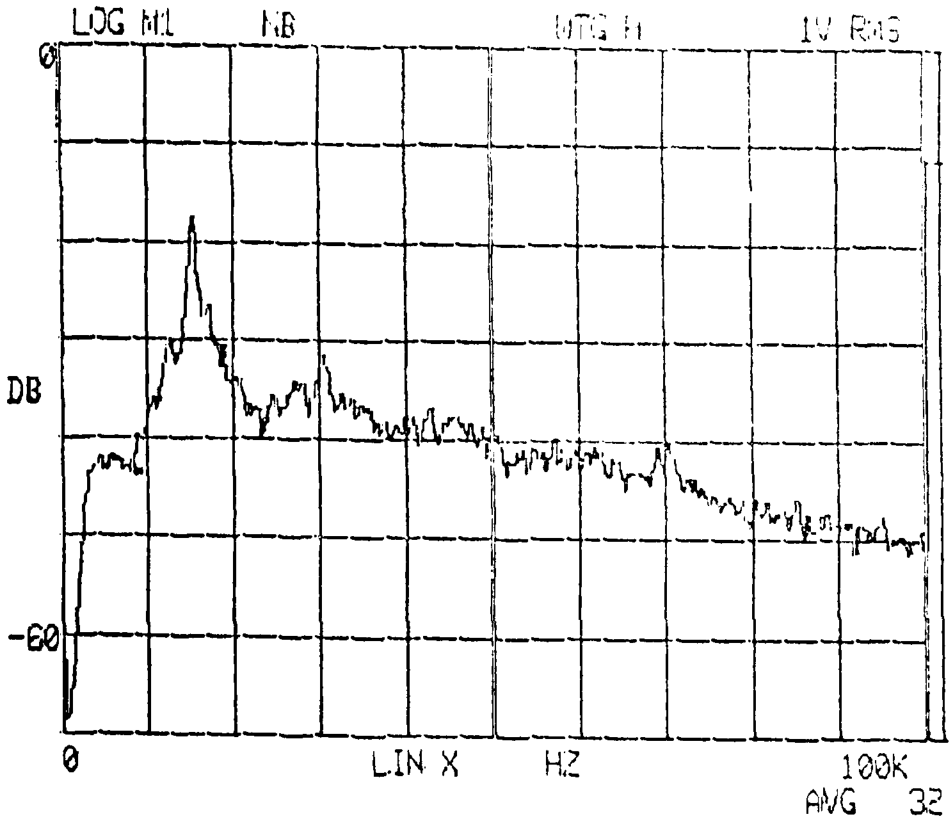


Figure 5.6 b: Frequency Spectrum (0-100 KHz) of Piston Pump. Cavitation Test, Inlet Pressure = 1.38 atm (Incipient)

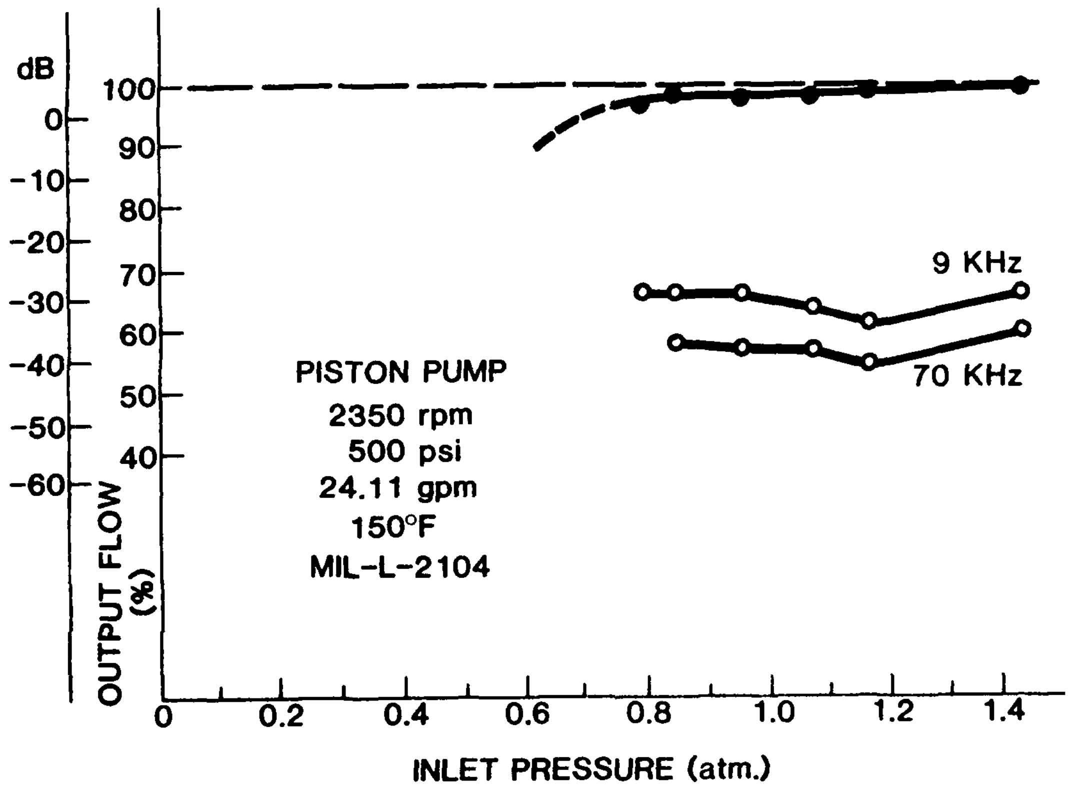


Figure 5.7: A.E. Signal Level VS. Pump Inlet Pressure from Piston Pump Cavitation Tests

The tests were performed on the test stand and also on the front loader. The results of these tests are presented in the next section.

5.3 Gear Pumps

Several experiments were conducted in this series of tests to determine the possible application of acoustic emission in the diagnostic evaluation of gear pumps. The following tests were performed:

(1) Cavitation

(2) Pump degradation

(3) Internal mechanical damage

A one-section pump and a two-section pump were evaluated on the pump test stand. Two identical two-section pumps were also tested on the front-end loader. Figure 5.8 shows the one-section pump. Figure 5.9 shows the two-section pump disassembled. Specifications for both types of pumps are given in Appendix B.

In the following sections, selected experimental results are presented and discussed. The problems encountered in using A.E. for the diagnostic evaluation of gear pumps are also discussed.

5.3.1 Incipient Cavitation

A one-section gear pump was tested on the pump test stand to determine if incipient cavitation in gear pumps could be detected using A.E. signature analysis. For this set of tests, the fluid temperature was held constant at 125^{o}F, and the outlet pressure was held constant at 500 psig. The pump was driven at 1800 RPM.

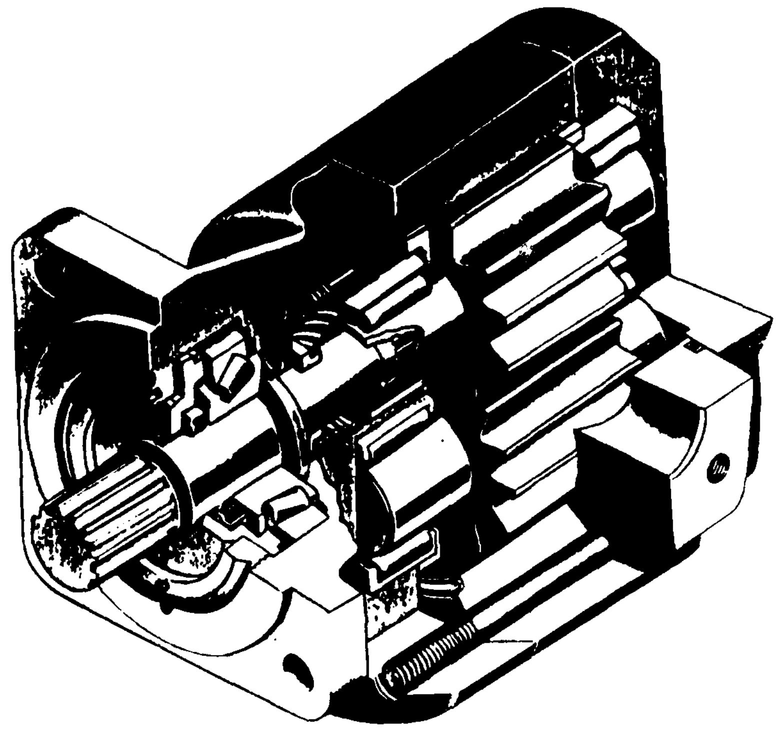


Figure 5.8: Cutaway View of One-Section Gear Pump

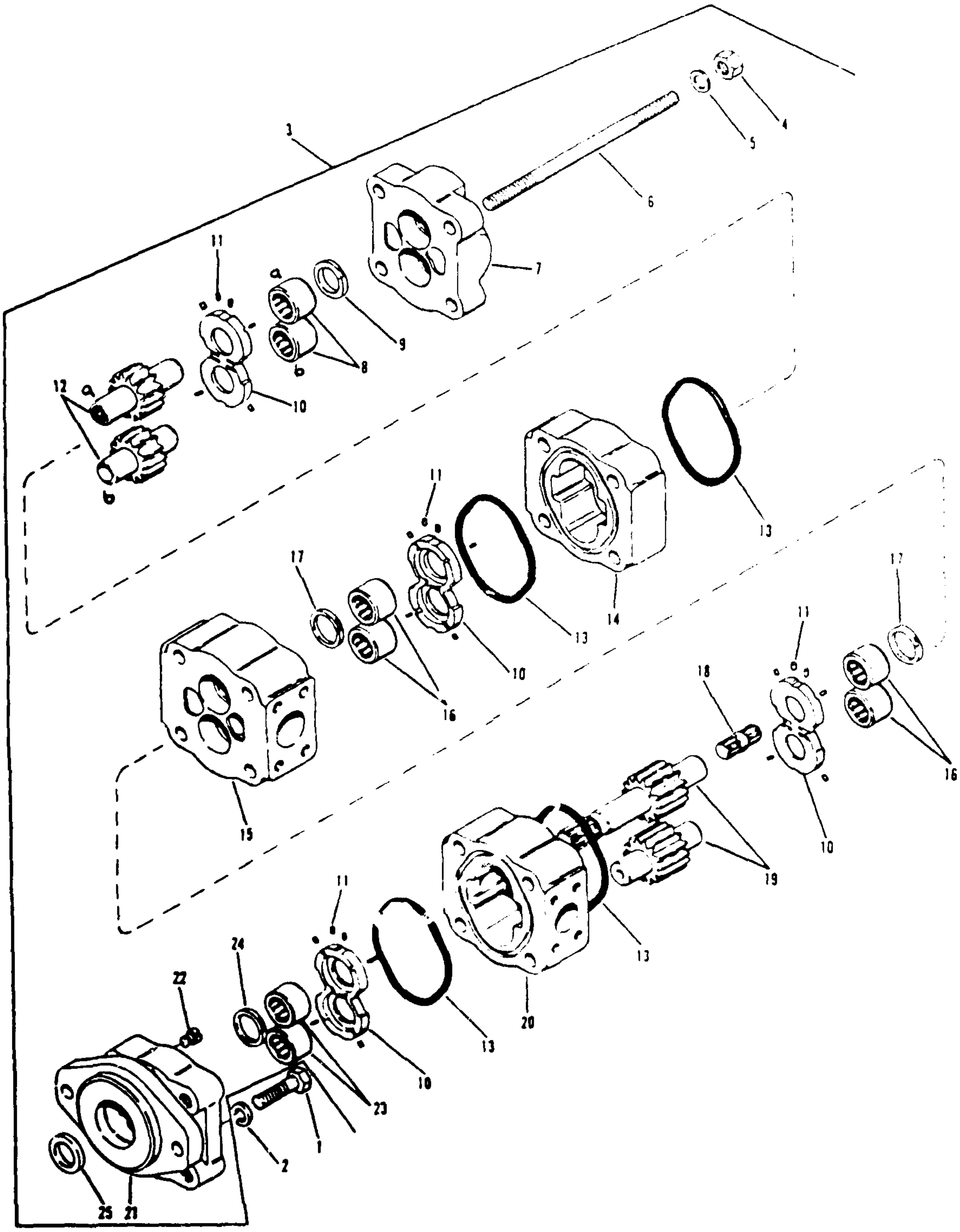


Figure 5.9: View of Disassembled Two-Section Gear Pump from Front Loader

The AC-75L transducer was used for these experiments. It was cemented to the housing of the pump. The transducer output was amplified using the AVLD and analyzed using a spectrum analyzer.

Figure 5.10 contains two spectra obtained from these tests. The spectrum in 5.10 a corresponds to the normal operating condition. The spectrum in 5.10 b corresponds to a measurable cavitation condition. As can be seen from this figure, the overall energy content of the signal is increased.

Expecting results similar to those obtained in the piston pump tests, the amplitude of various frequency regions was monitored for decreasing inlet pressure. Figure 5.11 shows the results of these tests. As in the piston pump results, the A.E. level initially decreases to a minimum. This minimum corresponds to the incipient cavitation point. Decreasing the inlet pressure past this point results in an increase in the A.E. amplitude. As can be seen in this figure, the amplitude at 70 KHz was measured at the various inlet pressures. In addition, the average signal level between 70 to 90 KHz and 90 to 100 KHz was monitored. All three of these variables (70 KHz, 70 to 90 KHz, or 90 to 100 KHz) could be used as an indicator of incipient cavitation.

Identical tests were performed on one of the two-section pumps. During these tests, the 0 - 100 KHz narrow band spectrum of the A.E. signal was obtained. In addition to this, the narrow-band spectrum from 28,250 Hz to 33,250 Hz was taken using the "zoom" function on the spectrum analyzer.

Figure 5.12 contains the spectra obtained from the two-section

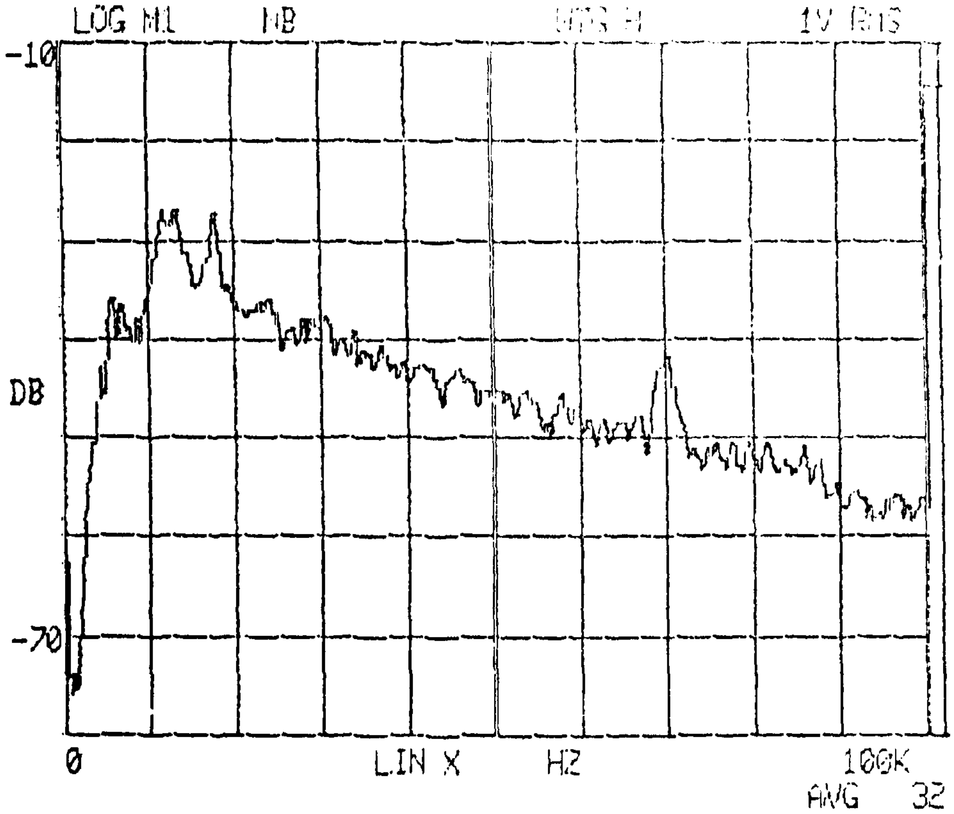


Figure 5.10 a: Frequency Spectrum of Gear Pump. Cavitation Test, Inlet Pressure = 1.27 Atm (0-100 KHz)

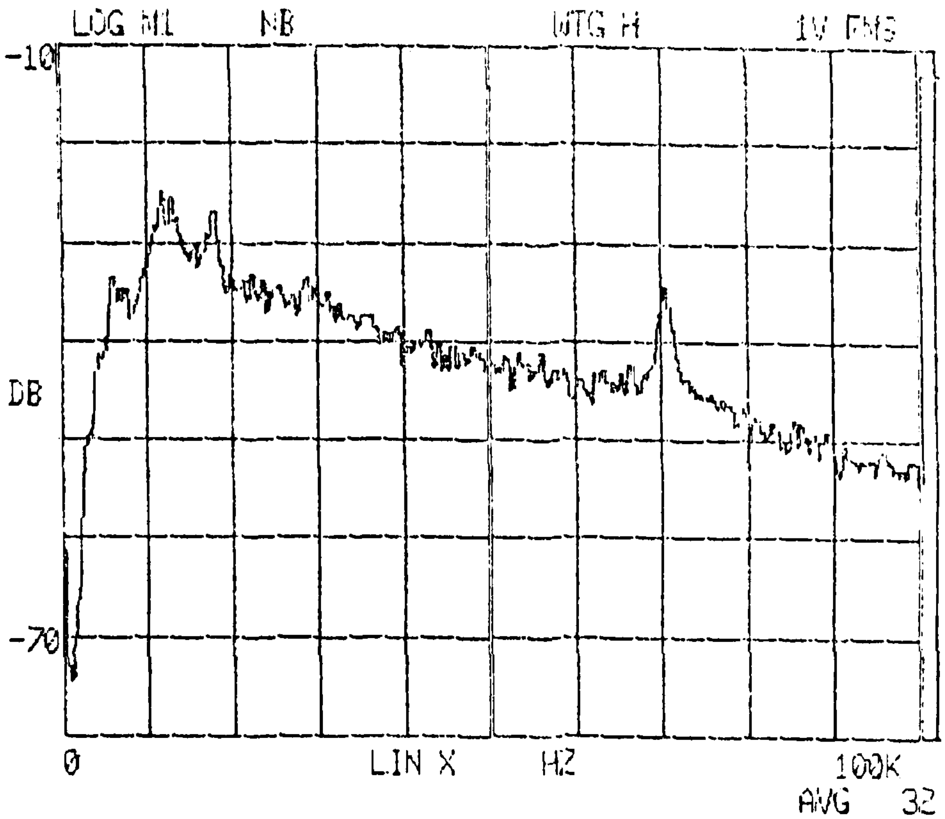


Figure 5.10 b: Frequency Spectrum of Gear Pump. Cavitation Test. Inlet Pressure = 0.59 Atm (0-100 KHz)

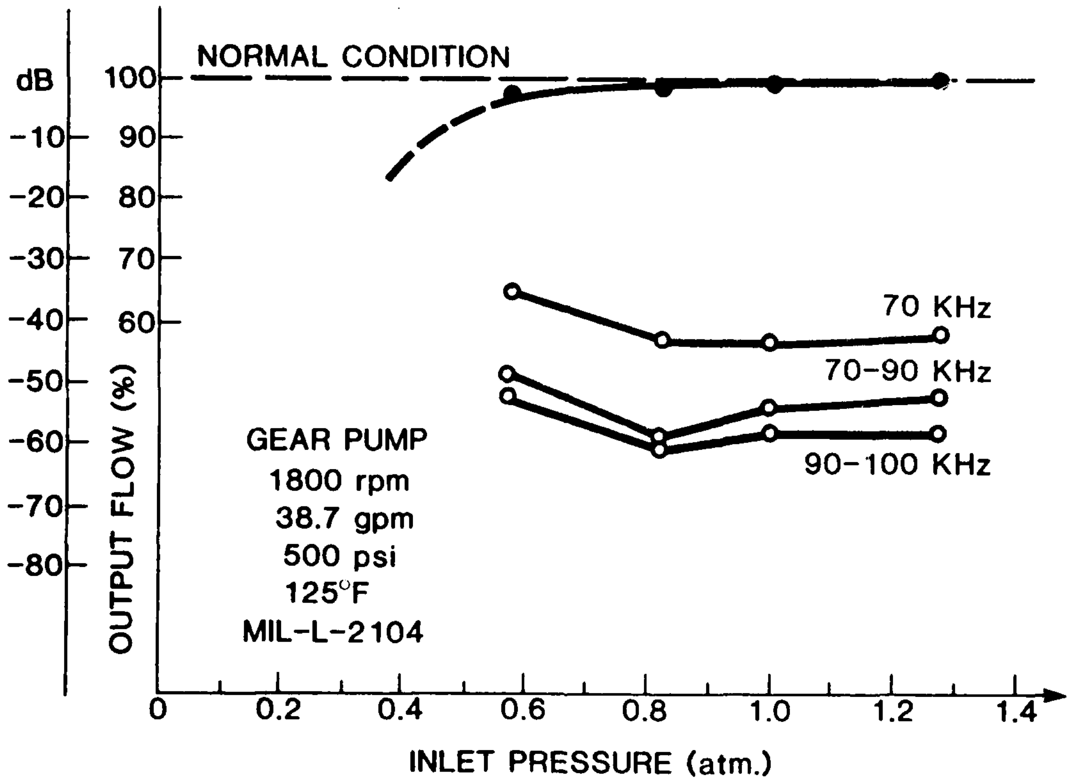


Figure 5.11: A.E. Signal Level VS. Pump Inlet Pressure Cavitation Tests of Gear Pump

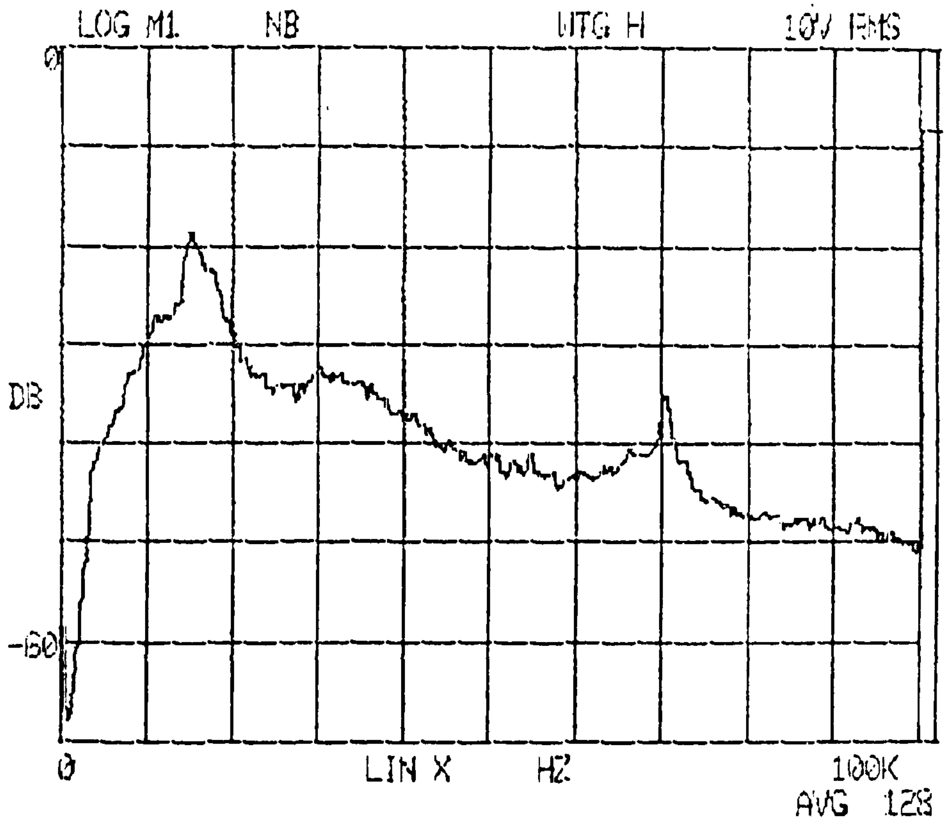


Figure 5.12 a: Frequency Spectrum of Two-Section Gear Pump Cavitation Test, Inlet Pressure = 1.34 Atm (0-100 KHz)

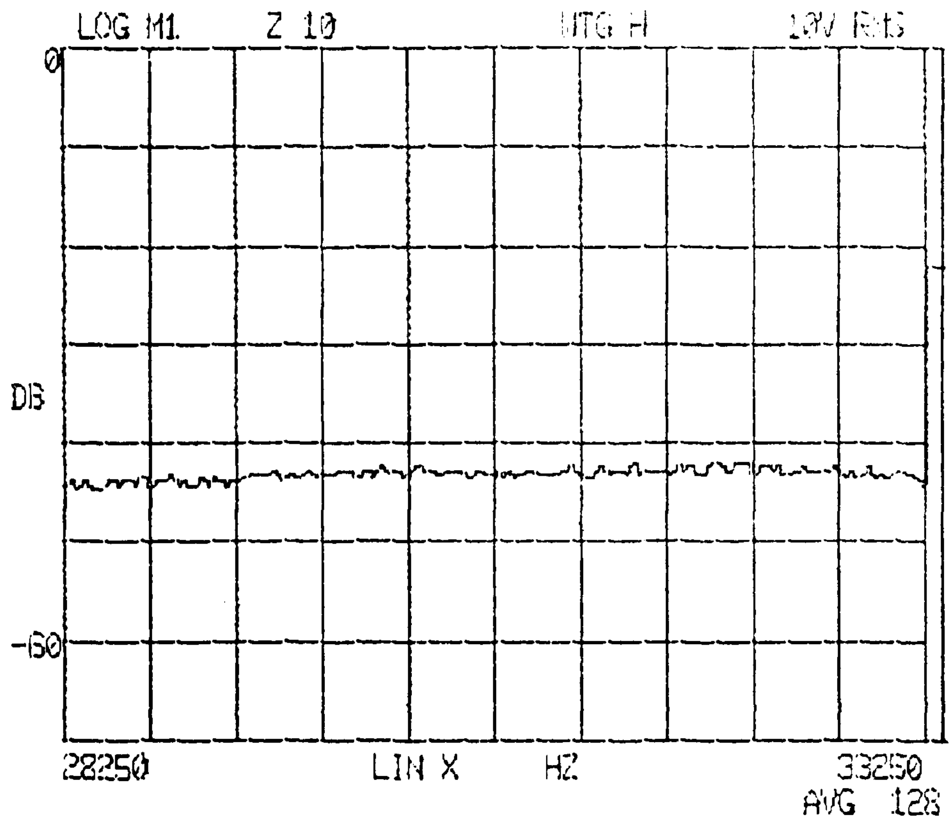


Figure 5.12 b: Zoom Frequency Spectrum of Two-Section Gear Pump Cavitation Test, Inlet Pressure = 1.34 Atm (28.25 - 33.25 KHz)

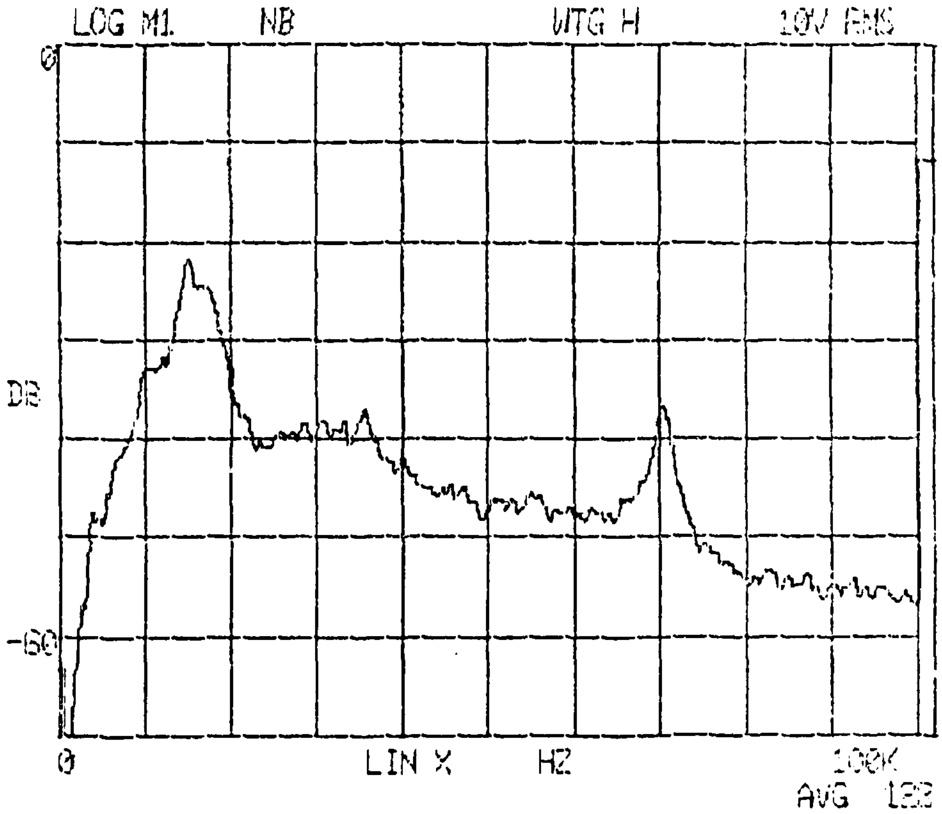


Figure 5.13 a: Frequency Spectrum of Two-Section Gear Pump
Cavitation Test, Inlet Pressure = 0.54 Atm
(0-100 KHz)

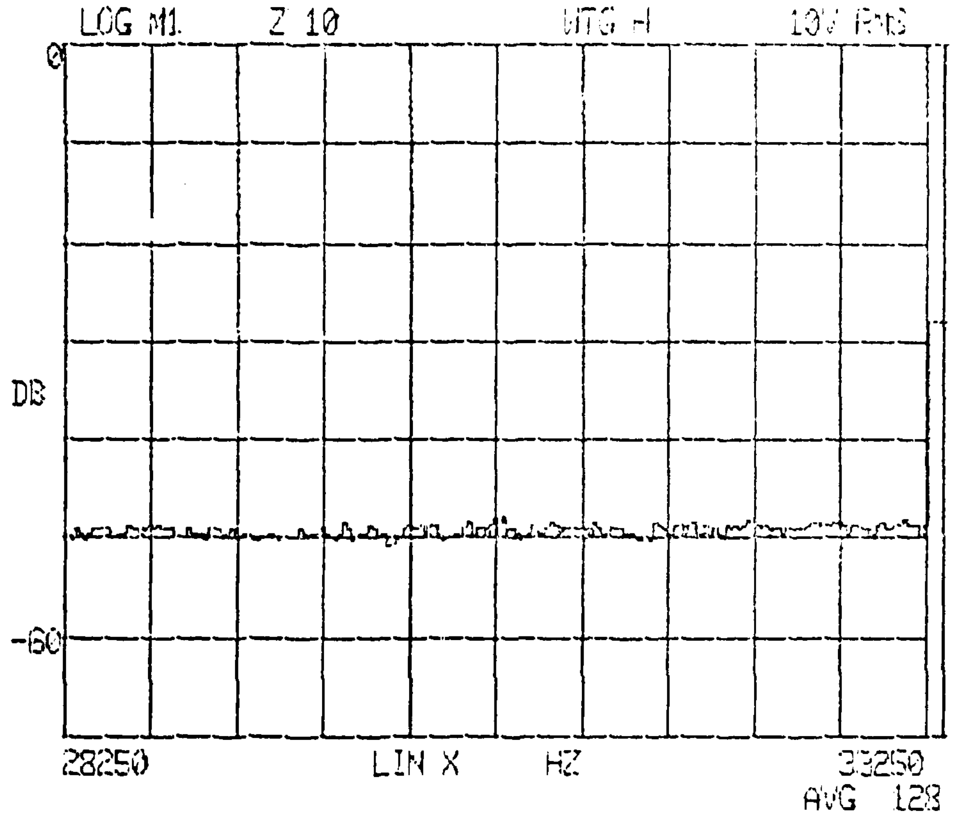


Figure 5.13 b: Zoom Frequency Spectrum of Two-Section Gear Pump Cavitation Test, Inlet Pressure - 0.54 Atm (28.25 - 33.25 KHz)

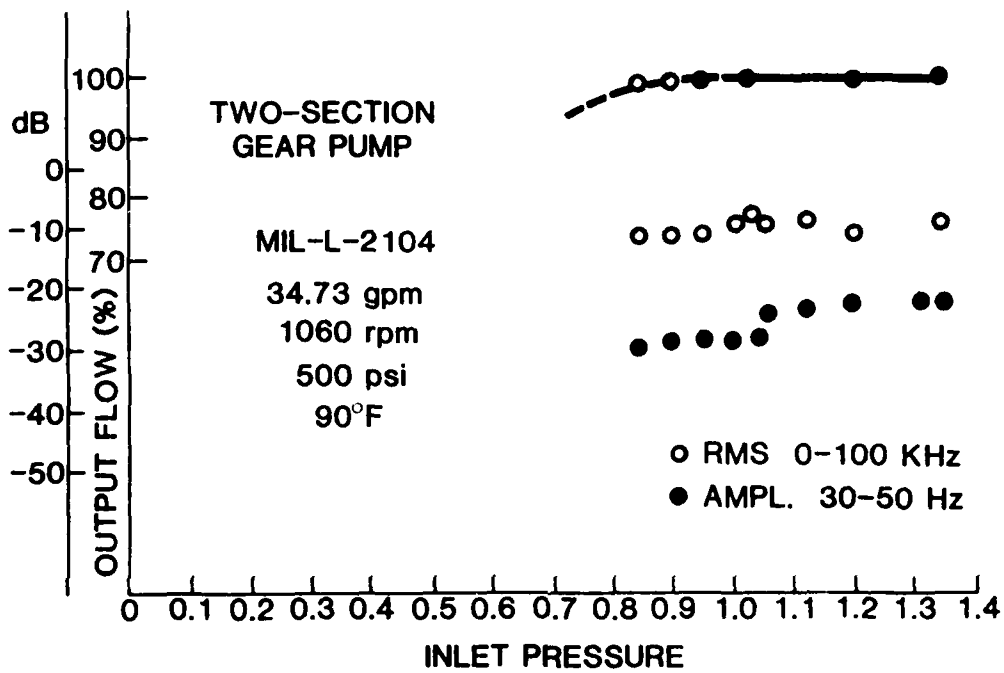


Figure 5.14: A.E. Signal Level VS. Pump Inlet Pressure for Two Section Pump - Cavitation Tests

pump under normal operating conditions. Figure 5.13 contains the spectra obtained when the pump was at the incipient cavitation point. As can be seen from these figures, the incipient cavitation point shows a measurable decrease in the A.E. signal level compared to the normal operating condition. Figure 5.14 contains the plots of the RMS of the A.E. signal versus inlet pressure and also the amplitude at 30,750 Hz versus inlet pressure. Both of these variables indicate the point of incipient cavitation but are not as sensitive as the other variables investigated.

In conclusion, incipient cavitation can be detected both in the one-section and the two-section pumps. Parameters such as the amplitude at 70 KHz or the average signal level between 70 to 90 KHz could be used as an indicator of incipient cavitation.

5.3.2 Degradation Tests

This set of tests involved the degradation of a one-section gear pump using standard contaminants. Figure 5.15 shows the decrease in output flow versus the contaminant size injected. The A.E. signal produced by the pump was recorded for the various levels of degradation at two different outlet pressures, 200 psig and 1500 psig. The shaft speed for these tests was held constant at 30 Hz.

The Ac-75L A.E. transducer was used for these tests. Its output was amplified using the AVLD and analyzed using a spectrum analyzer.

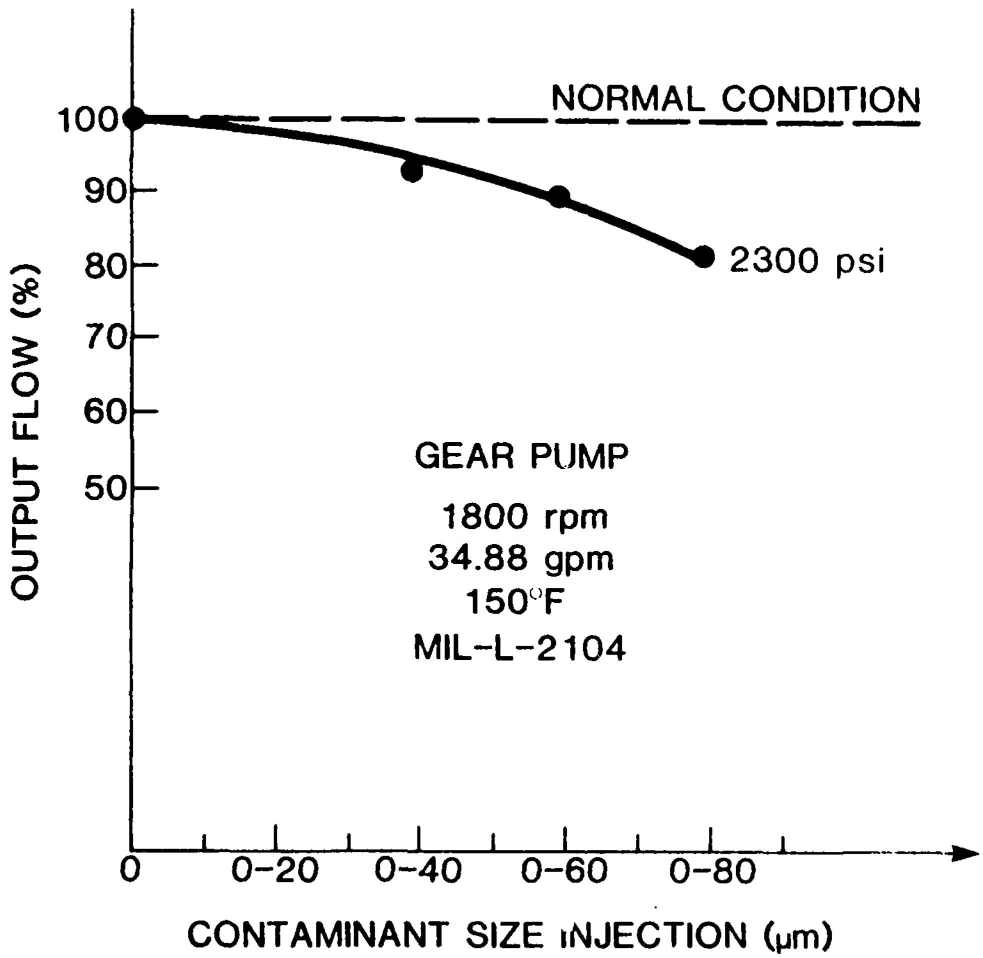


Figure 5.15: Pump Output Flow Degradation VS. Contaminant Size - Gear Pump

Figures 5.16, 5.17, 5.18, and 5.19 contain the spectra obtained at the two different outlet conditions and the various degradation levels. From these spectra, Figure 5.20 was constructed. This figure contains the plot of the amplitude at 60.0 KHz of the A.E. signal versus the pump degradation level (contaminant size injected) for the two pressure conditions. This figure also contains the same type plot for the amplitude at 6.8 KHz and the two pressure conditions.

Figure 5.20 shows that initial degradation of the pump caused a decrease in the A.E. signal level for the 200 psig outlet condition. Continued degradation caused the A.E. signal to reach a minimum level. Further degradation beyond this point resulted in an increase in the A.E. signal level. The amplitude at 60 KHz for the 100 psig outlet condition clearly shows this variation.

In conclusion, degradation of a gear pump can be detected from the A.E. signal. For the low pressure outlet condition, the A.E. signal level initially decreases with increasing degradation level until a minimum level is reached. Degradation of the pump beyond this point causes an increase in the A.E. signal level. For this technique to be a "wear-out" indicator, it is likely that the spectra for each pump would have to be monitored throughout its life-span.

5.3.3 Internal Mechanical Damage Test Procedure

The purpose of these tests was to determine the types of internal mechanical damage that could be detected in the A.E. signal of a pump.

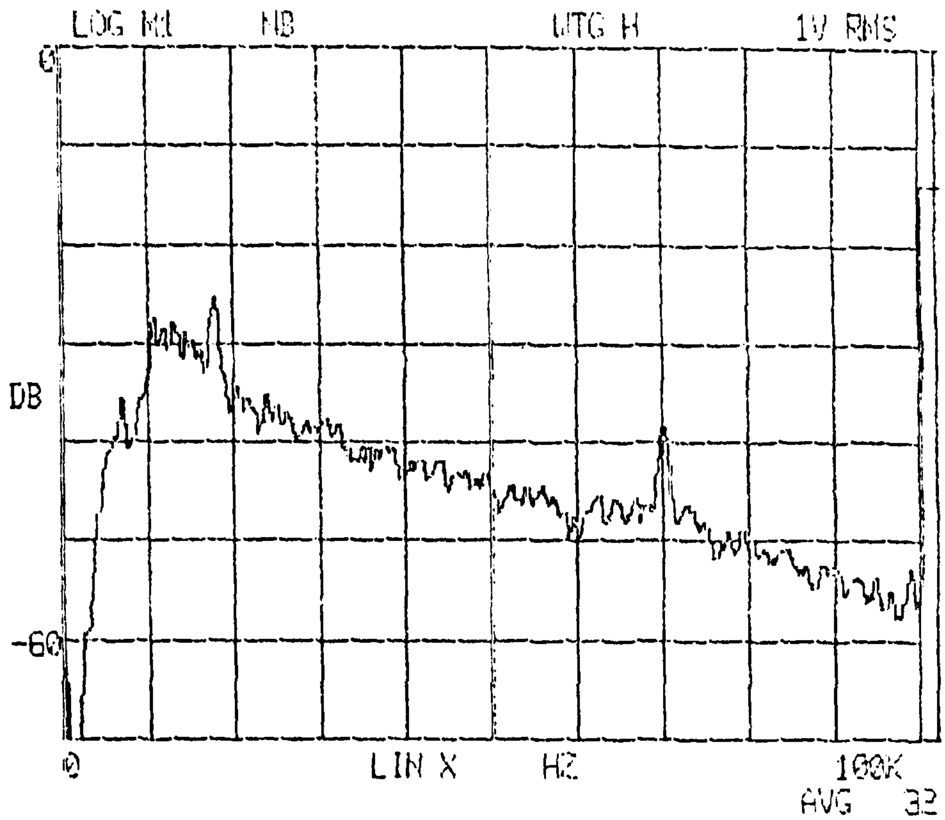


Figure 5.16 a: Gear Pump Frequency Spectrum (0-100 KHz)
Degradation Test. 0% Degradation, Outlet
Pressure = 200 psig

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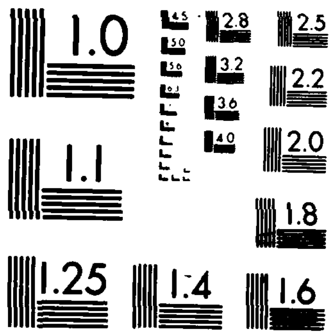

MICROCOPY RESOLUTION TEST CHART
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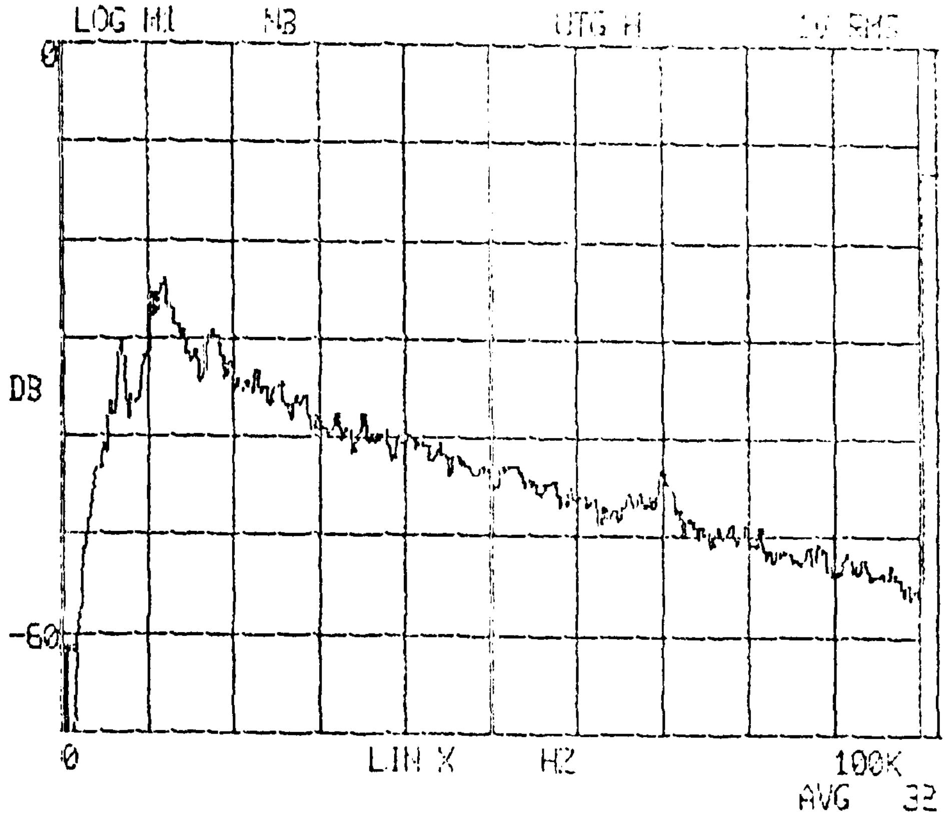


Figure 5.16 b: Gear Pump Frequency Spectrum (0-100 KHz) Degradation Test, 0% Degradation, Outlet Pressure = 1500 psig

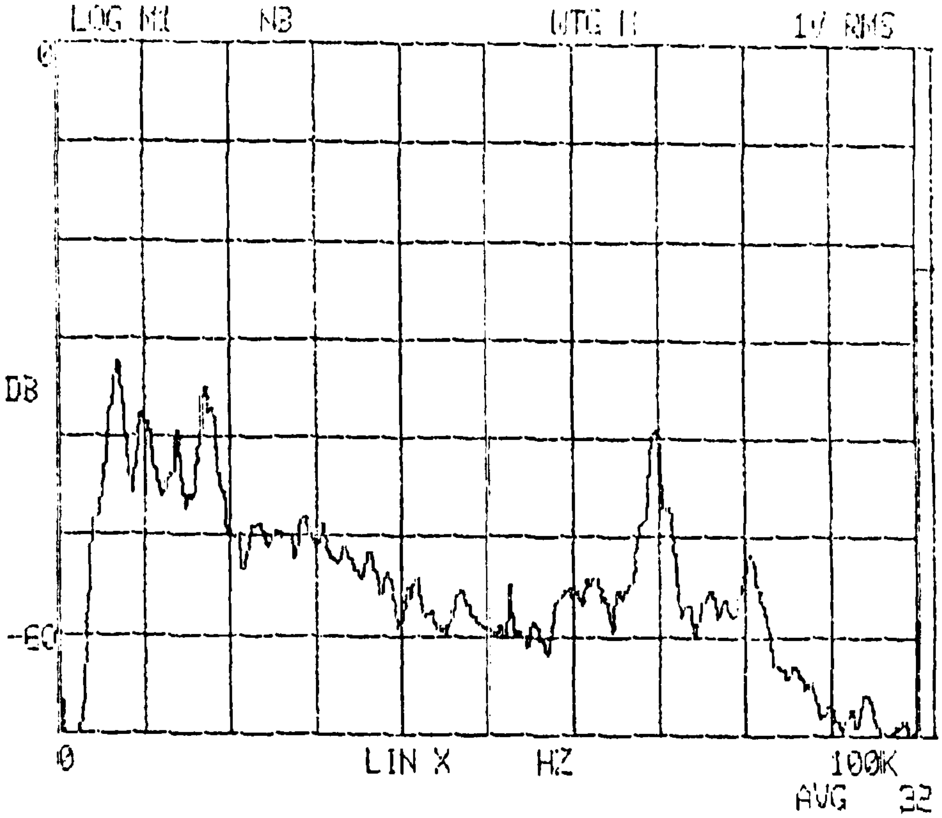


Figure 5.17 a: Gear Pump Frequency Spectrum (0-100 KHz)
Degradation Test, 6.6% Degradation Outlet
Pressure = 200 psig

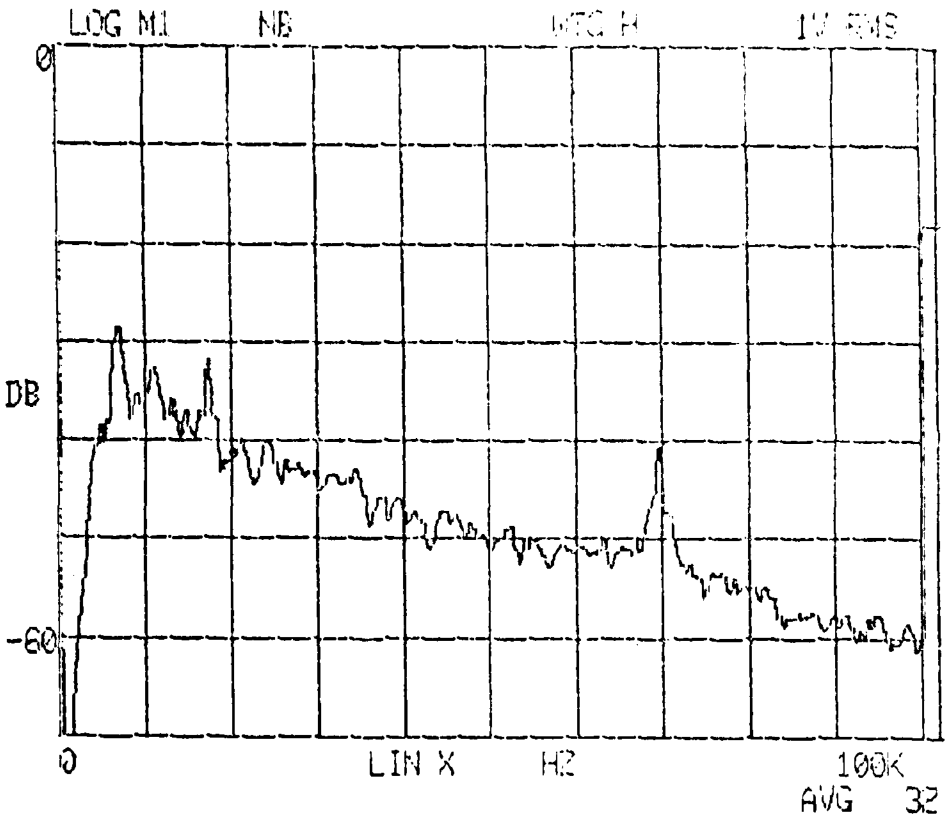


Figure 5.17 b: Gear Pump Frequency Spectrum (0-100 KHz)
Degradation Test, 6.6% Degradation Outlet
Pressure = 1500 psig

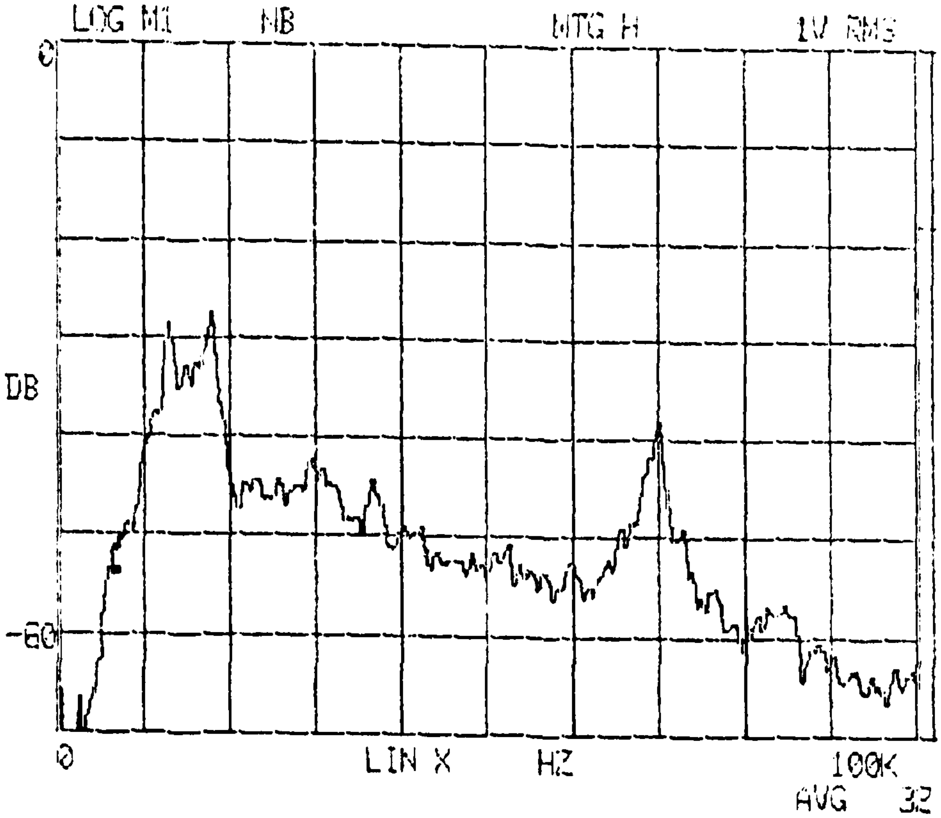


Figure 5.18 a: Gear Pump Frequency Spectrum (0-100 KHz)
Degradation Test, 12% Degradation Outlet
Pressure = 200 psig

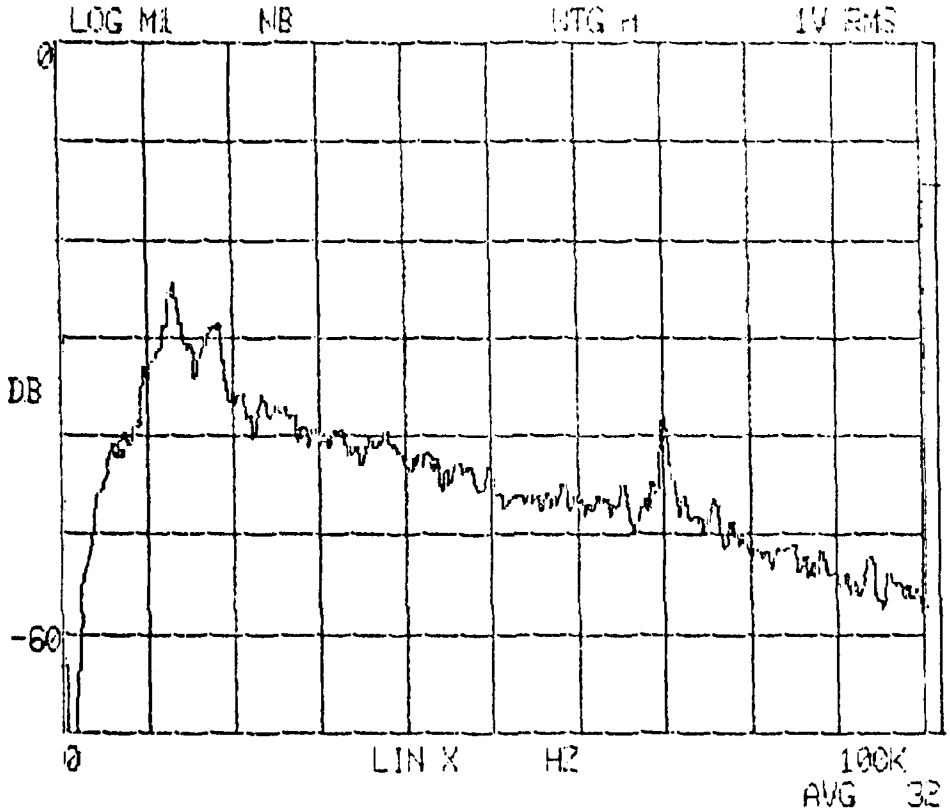


Figure 5.18 b: Gear Pump Frequency Spectrum (0-100 KHz)
Degradation Test, 12 Degradation Outlet
Pressure = 1500 psig

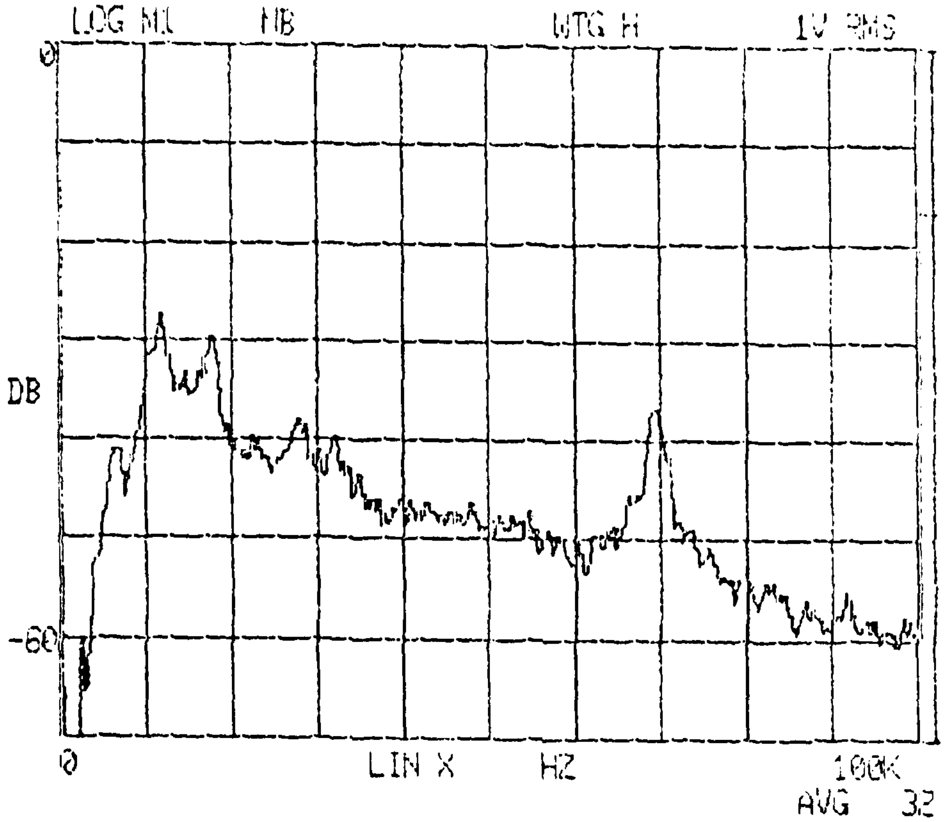


Figure 5.19 a: Gear Pump Frequency Spectrum (0-100 KHz)
Degradation Test, 20% Degradation Outlet
Pressure = 200 psig

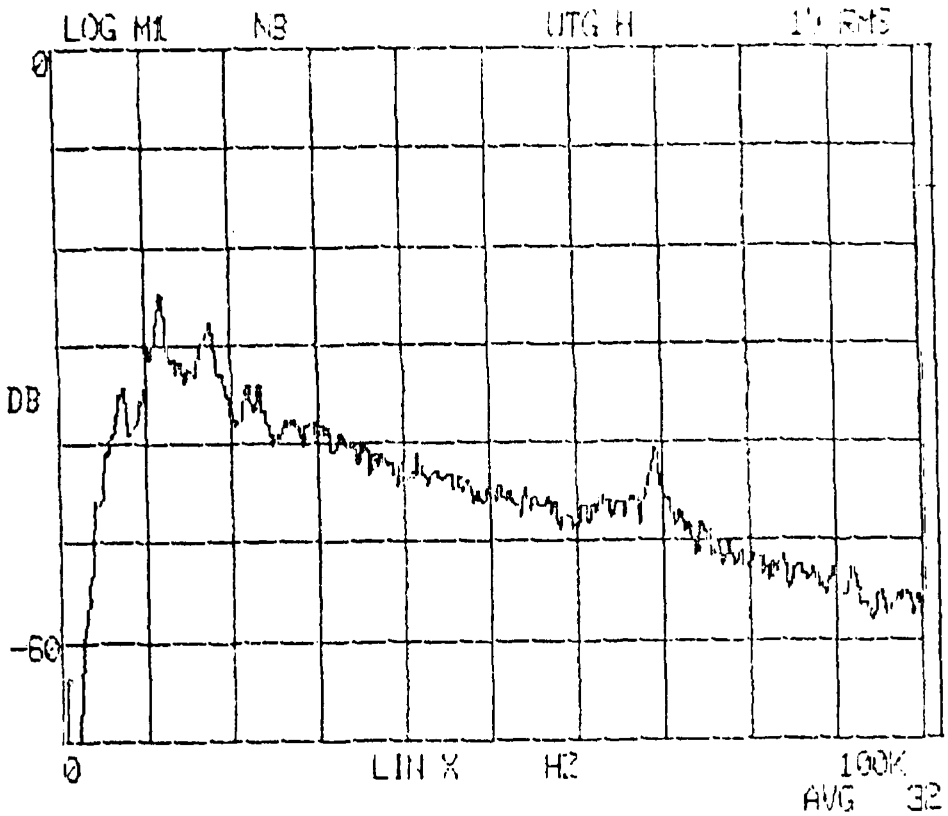


Figure 5.19 b: Gear Pump Frequency Spectrum (0-100 KHz)
Degradation Test, 20% Degradation Outlet
Pressure = 1500 psig

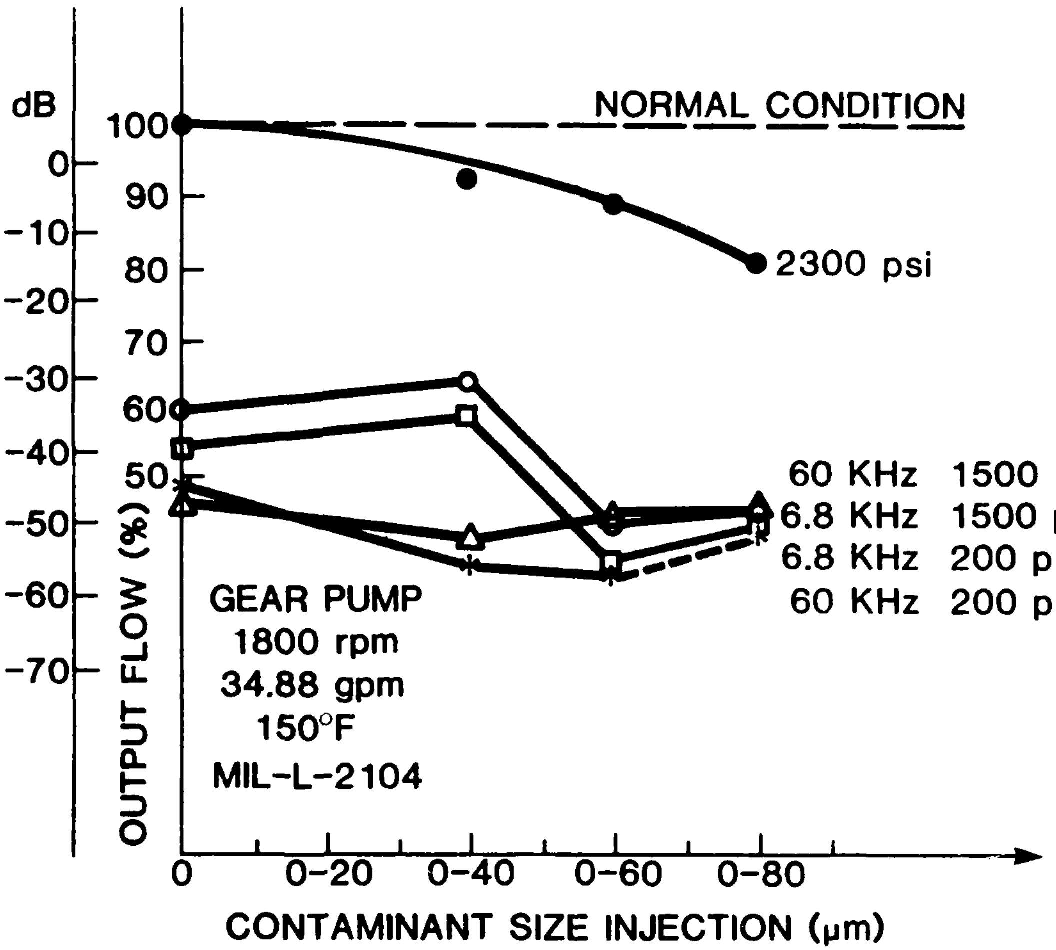


Figure 5.20: A.E. Signal Level VS. Pump Degradation - Gear Pump

The damage types investigated were: (1) damage to the shaft of the pump and (2) damage to a pair of mating gear teeth. A one-section gear pump was used for this set of tests.

The pump test procedure involved operating the pump at a specific shaft speed, inlet pressure and outlet pressure while recording the A.E. signal produced. These readings were used as a baseline for comparison of subsequent A.E. measurements. After obtaining the baseline data, the pump was disassembled and damaged internally. The pump was then reassembled and operated at the same conditions of the baseline test. This was done to determine if the damage inflicted caused a measurable change in the A.E. signal.

The first set of tests was performed with a shaft of the pump having been damaged. The damage was such that the rollers of one bearing were made to run over a rough and uneven surface. For this set of tests, the pump outlet pressure was held at 500 psig and the shaft rotational speed was 1200 RPM. The A.E. transducer used was the Dunegan/Endevco S9204 transducer. Its output was amplified using the AVLD. The averaged narrow-band spectrum from 0 to 100 KHz of the A.E. signal was obtained using a spectrum analyzer.

The second set of tests was performed with a pair of mating gear teeth of the pump having been damaged. The impacting surfaces of the teeth were damaged so that the severity of the tooth impact was increased. The shaft rotational speed was held at 1200 RPM. The pump was tested at two outlet pressures -- 500 psig and 2000 psig. Several different A. E. transducers were used for this test. These include the AC-75, AC-175, and S9202. The output from these transducers was amplified using the AVLD. In addition to these A.E. transducers, an accelerometer was tested. Its output was amplified using a Kistler Charge Amplifier. The outputs from these transducers were analyzed in the frequency domain using a spectrum analyzer.

The final set of tests involved testing the gear pump with gear tooth damage once more. This time, however, the A.E. signal was analyzed in the time domain. The shaft speed was again 1200 RPM. The pump outlet pressure was held at 0 psig and the Dunegan/Endevco S9202 A.E. transducer was used. The unamplified output from the transducer was recorded on a Nicolet digital oscilloscope. The time record length was 80 msec. This provided enough time to obtain the signal variation for one revolution of the shaft (Period Shaft = 50 msec).

5.3.4 Results from the Mechanical Damage Tests

Many vibration signatures were observed and recorded during the pump tests. For reasons of clarity, however, only a limited number are presented. Figure 5.21 a-d contains two spectra obtained from the shaft damage tests. Figure 5.25-a contains the frequency spectrum obtained from the pump with severe shaft damage. It can be seen from these figures that no significant difference exists between the two spectra.

Figures 5.23-5.26 contain the spectra obtained from the gear tooth damage tests. Even with severe tooth damage, the frequency spectra shown

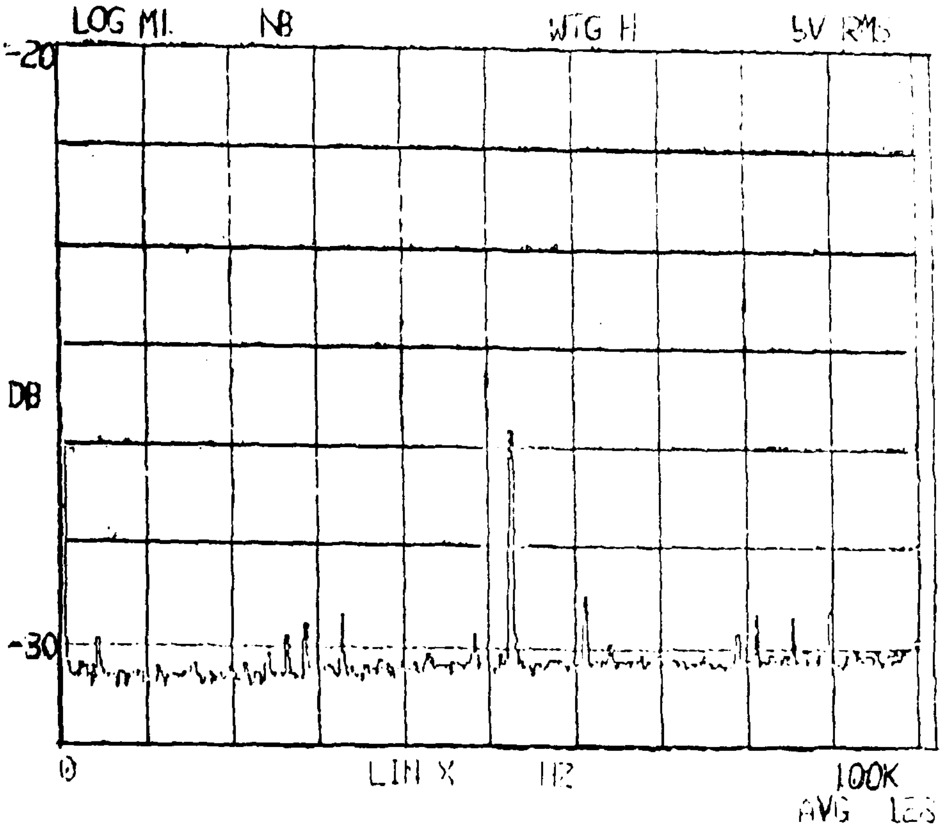


Figure 5.21 a: Background Signature of S9204 AA06 Transducer used on One-Section Gear Pump Tests

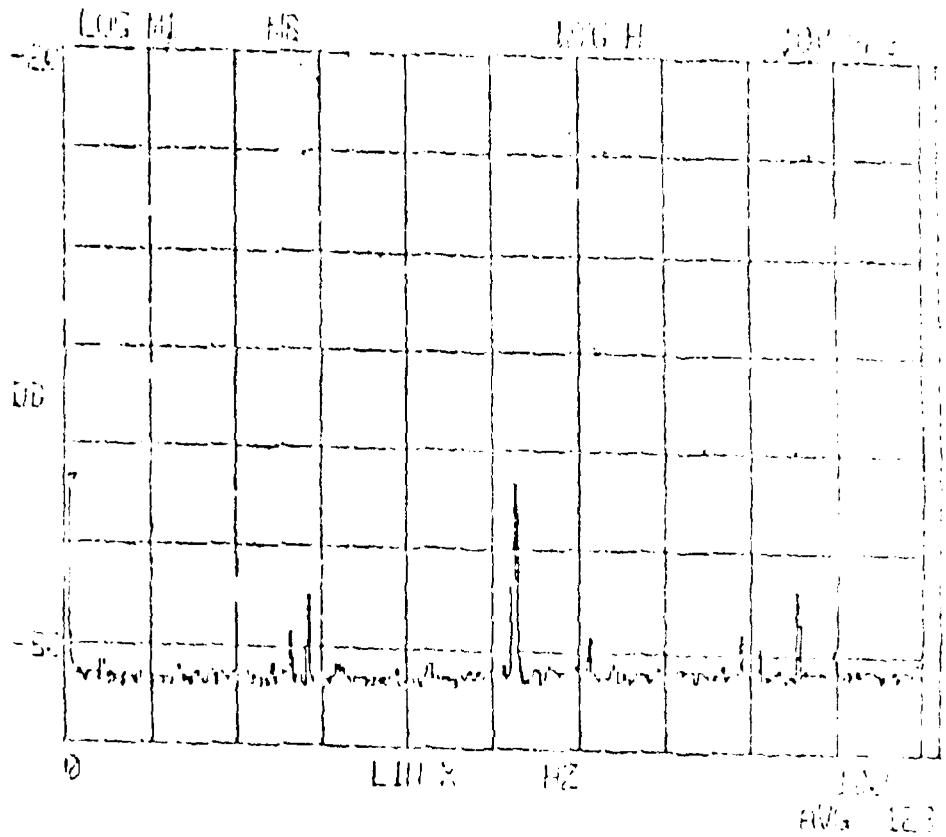


Figure 5.21 b: Background of Signature of AC - 75L Transducer used on One-Section Gear Pump Tests

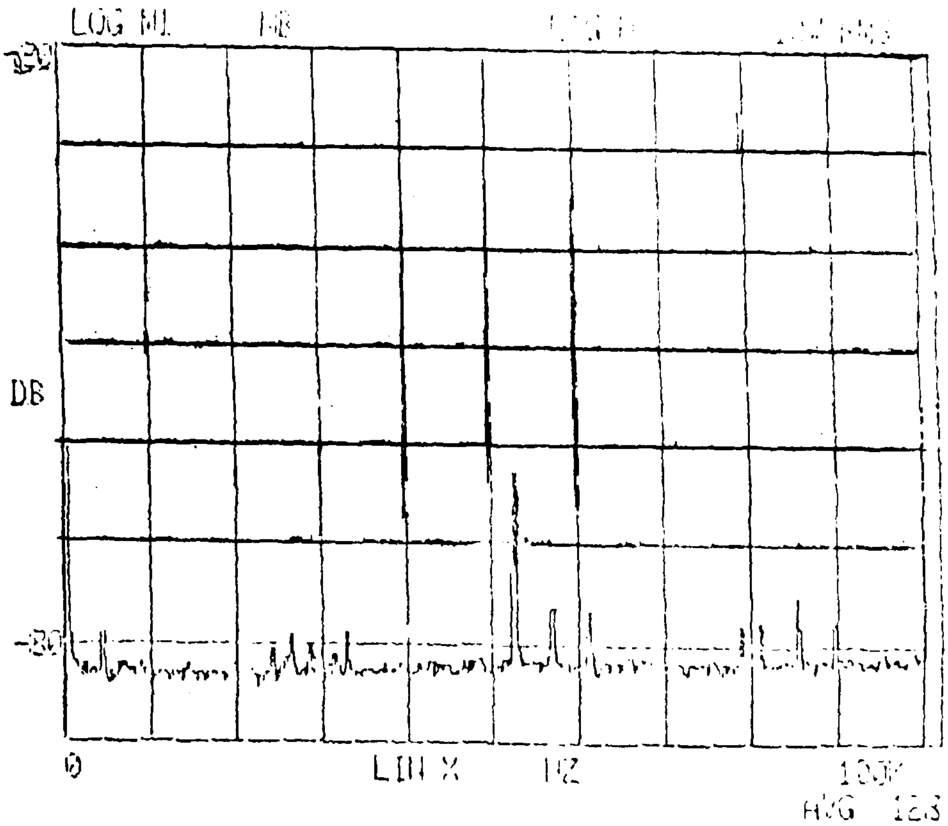


Figure 5.21 c: Background Signature of AC-175L Transducer used on One-Section Gear Pump Tests

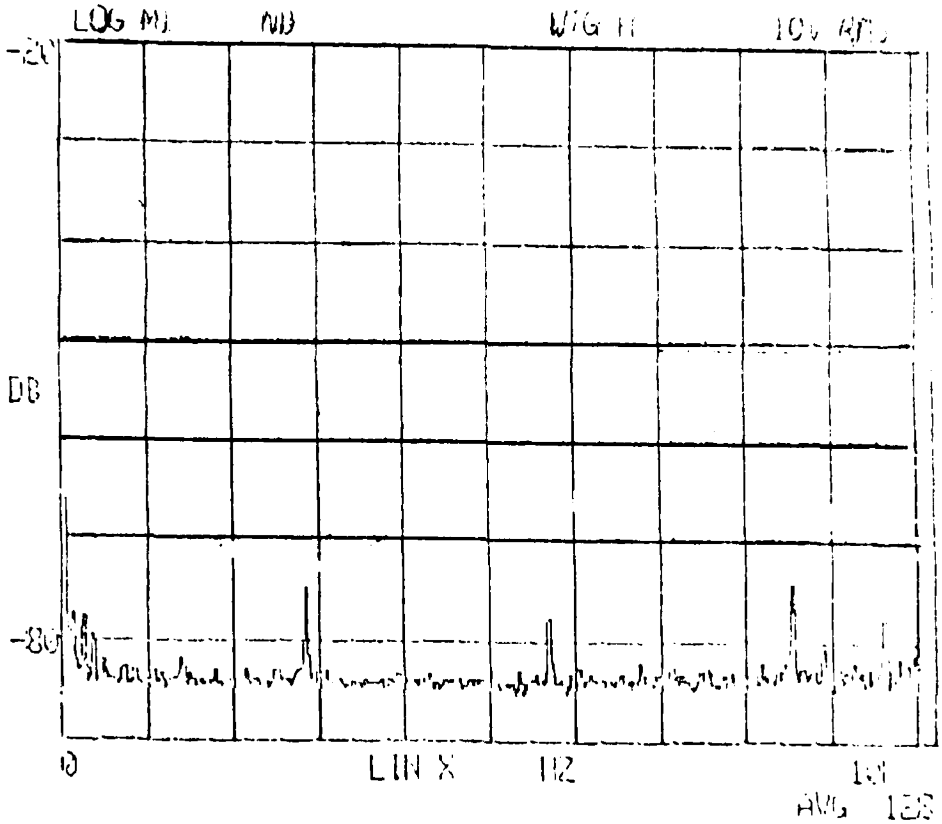


Figure 5.21 d: Background Signature of Accelerometer 3892 used on One-Section Gear Pump tests

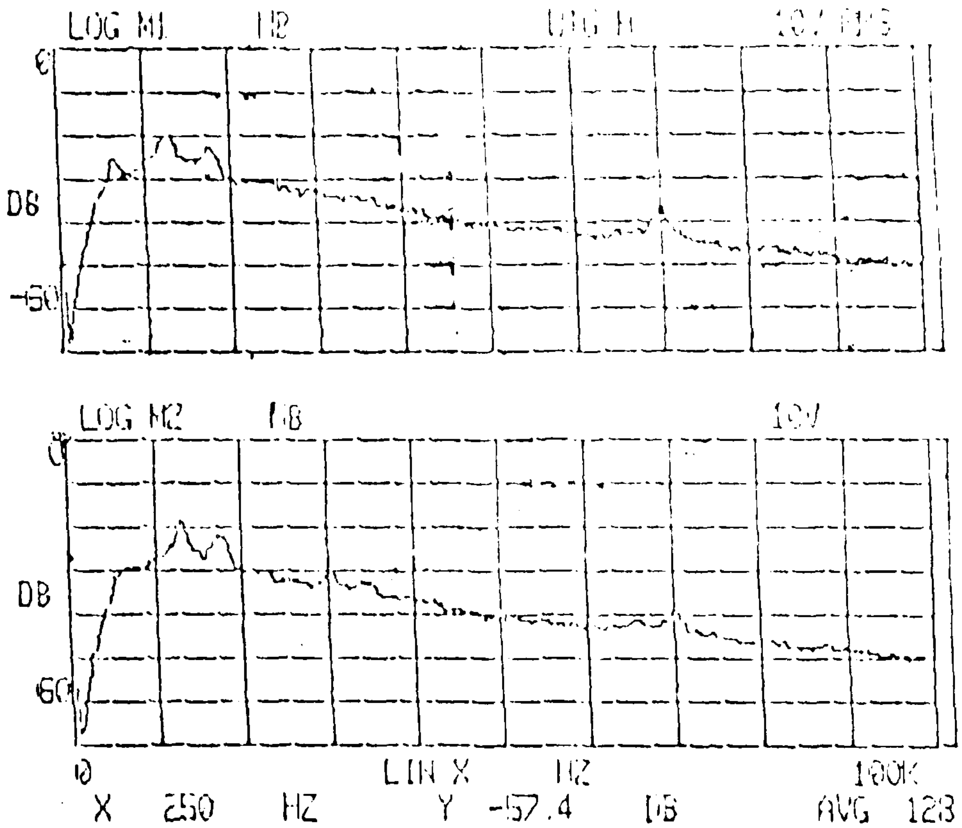


Figure 5.22 a: Frequency Spectrum of Gear Pump - Undamaged State (0-100 KHz) AC-75L Transducer

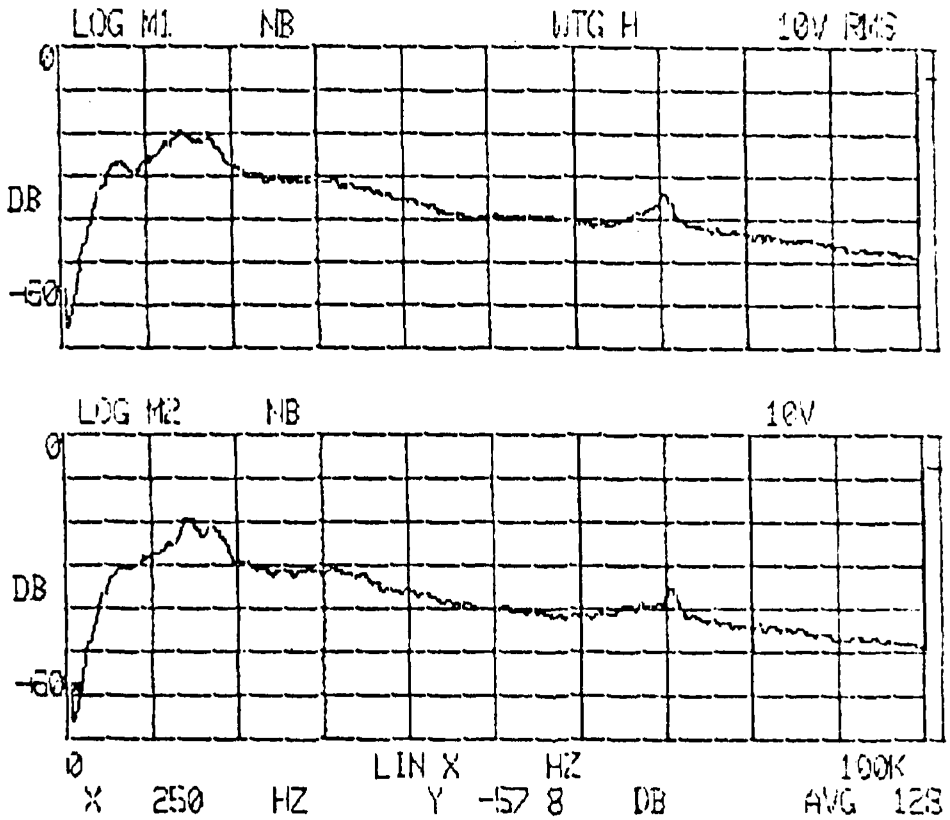


Figure 5.22 b: Frequency Spectrum of Gear Pump with Severe Tooth Damage (0-100 KHz) AC-75L Transducer

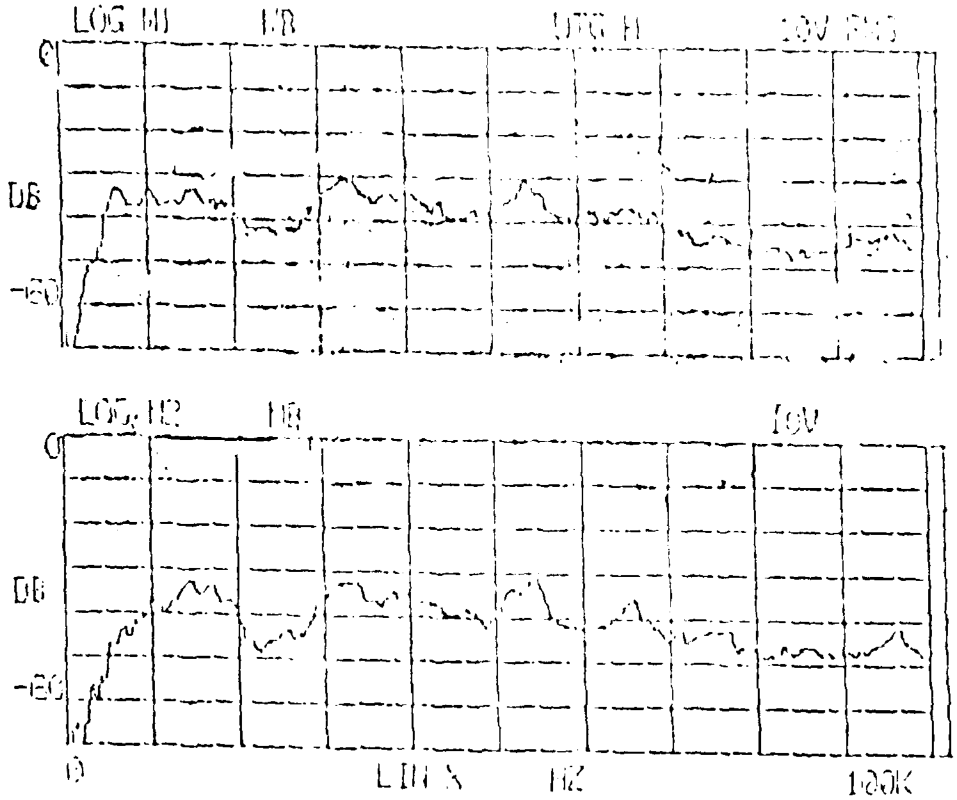


Figure 5.23 a: Frequency Spectrum of Gear Pump - Undamaged State (0-100 KHz) AC-175L Transducer

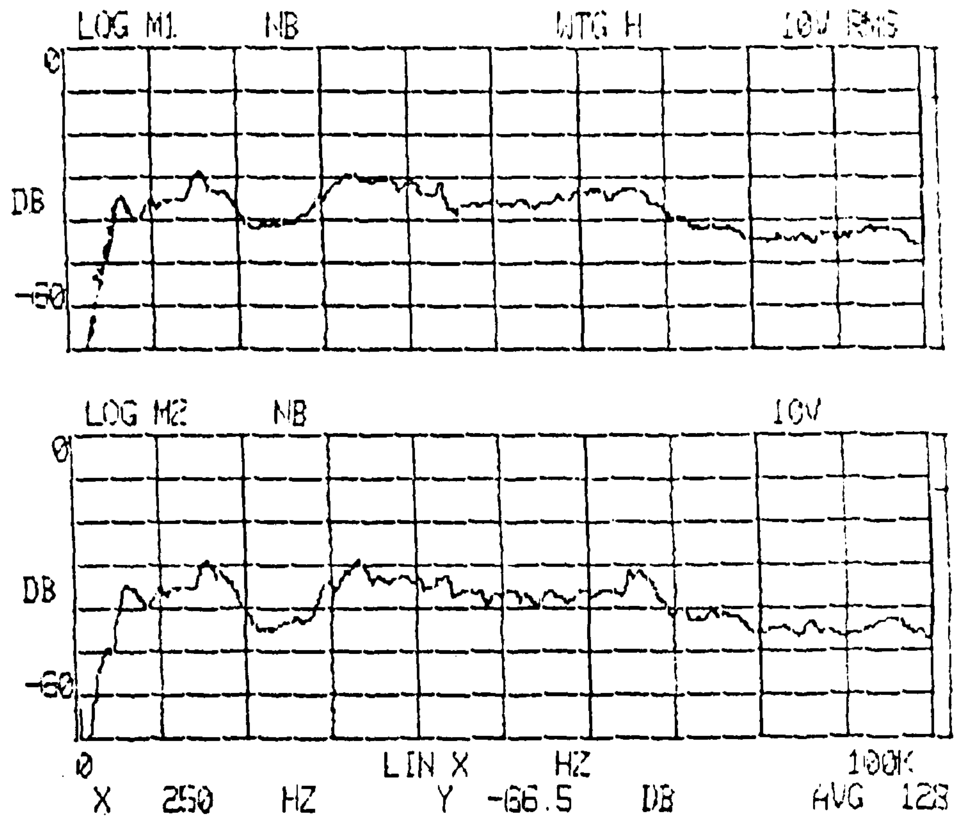


Figure 5.23 b: Frequency Spectrum of Gear Pump with Severe Tooth Damage (0-100 KHz) AC-175L Transducer

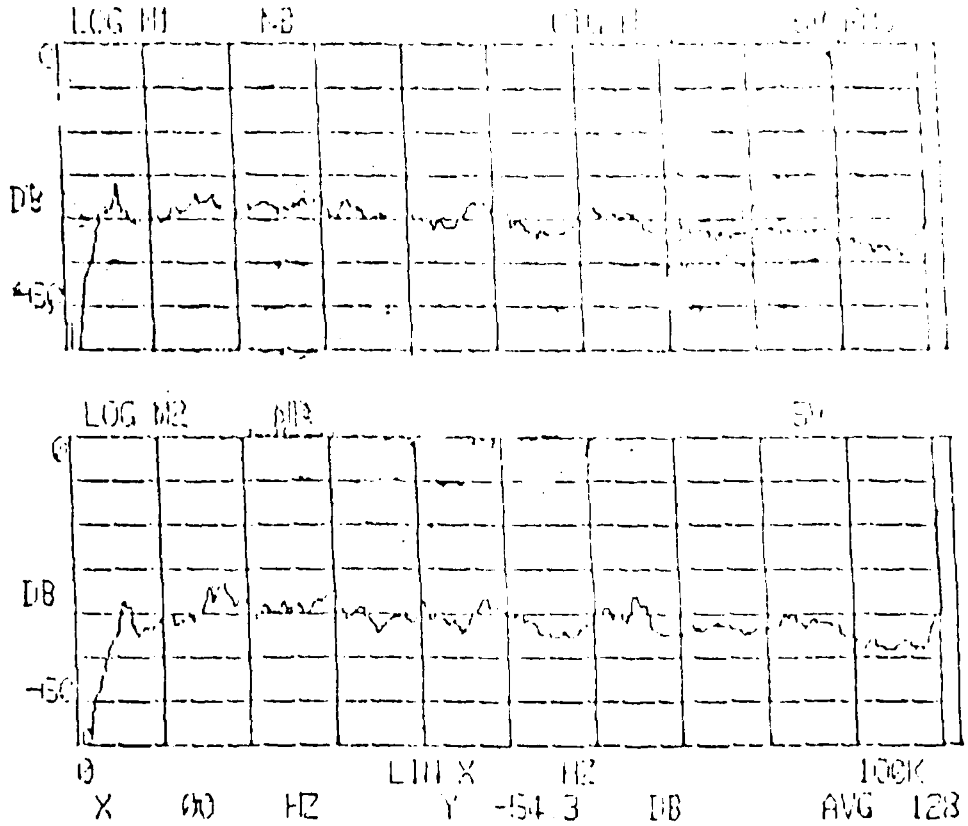


Figure 5.24 a: Frequency Spectrum of Gear Pump - Undamaged State (0-100 KHz) S9204 AA06 Transducer

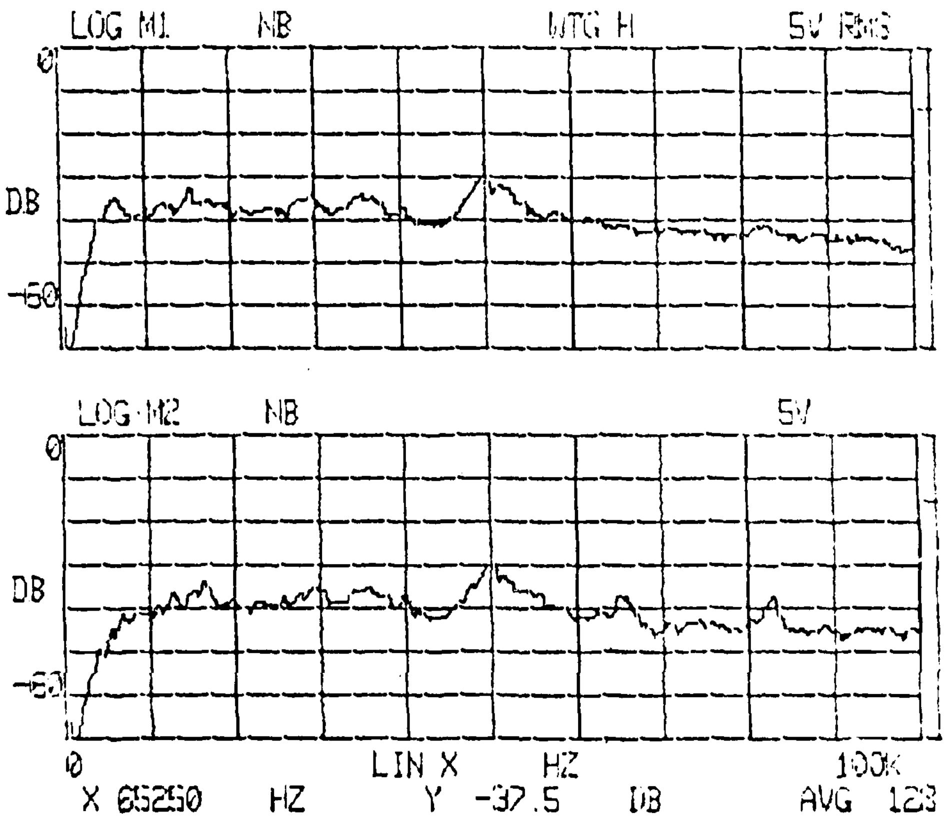


Figure 5.24 b: Frequency Spectrum of Gear Pump with Severe Tooth Damage (0-100 KHz) S 9204 AA06 Transducer

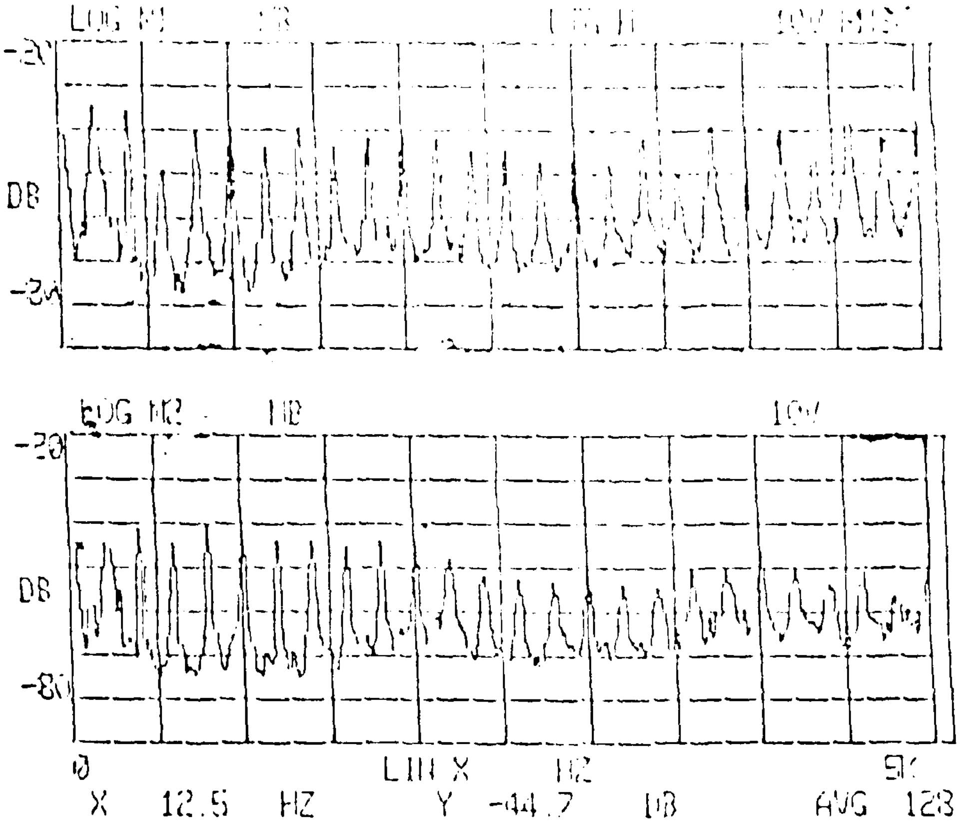


Figure 5.25 a: Frequency Spectrum of Gear Pump - Undamaged State (0-5 KHz) 3892 Accelerometer

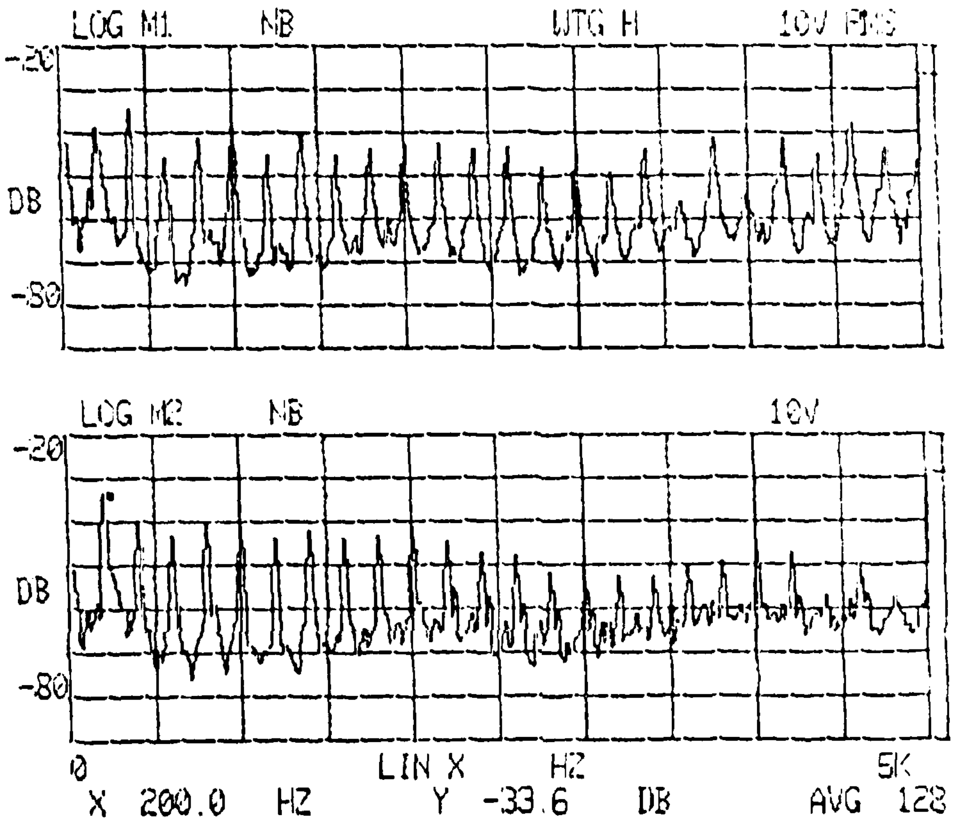


Figure 5.25 b: Frequency Spectrum of Gear Pump with Severe Tooth Damage (0-5 KHz) 3892 Accelerometer

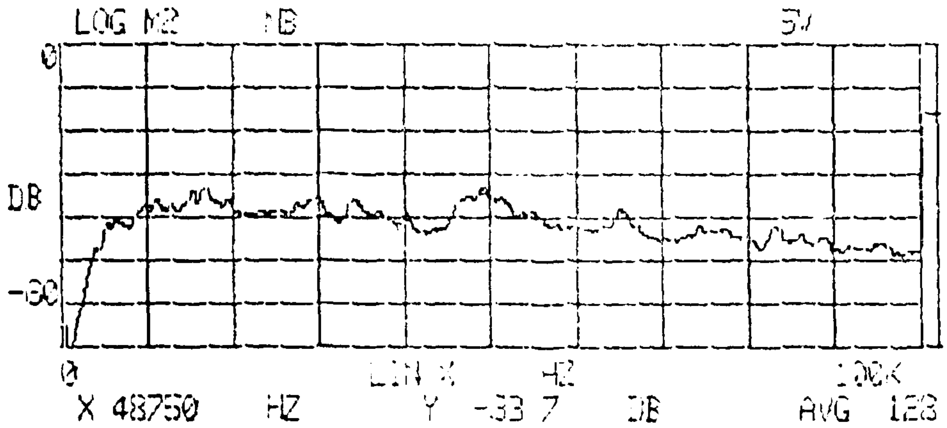


Figure 5.26 a: Frequency Spectrum of Undamaged Gear Pump (0-100 KHz) S9204 Transducer

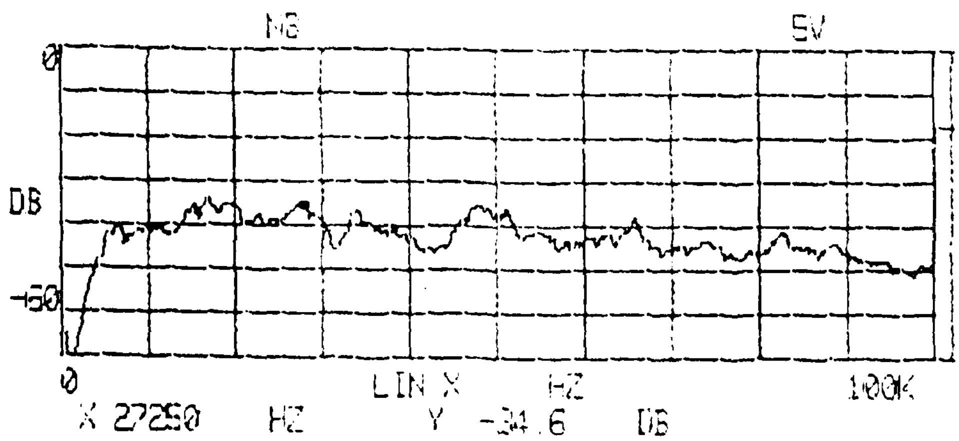


Figure 5.26 b: Frequency Spectrum of Gear Pump with Severe Shaft Damage (0-100 KHz) S 9204 Transducer

in these figures show no corresponding difference between the damaged and undamaged states. The failure of spectral analysis of the A.E. signal to provide diagnostic information on these tests prompted the addition of a time domain analysis to the gear tooth damage tests. The outlet pressure for the time domain tests was 0 psig, so that the high pressure flow noise would be minimized, allowing better measurement of the internal mechanical contact noise.

Figures 5.27 a-b contain the time records of the unamplified transducer output obtained during these tests. Figure 5.27-a corresponds to the undamaged pump, and Fig. 5.27-b corresponds to the undamaged pump. Figure 5.27-b shows that the damaged mating teeth cause the transducer output to "ring" from one tooth meshing to the next. This result shows that tooth damage can be detected in the time domain. These tests were repeated on the loader. The procedure and results obtained are discussed in the next section.

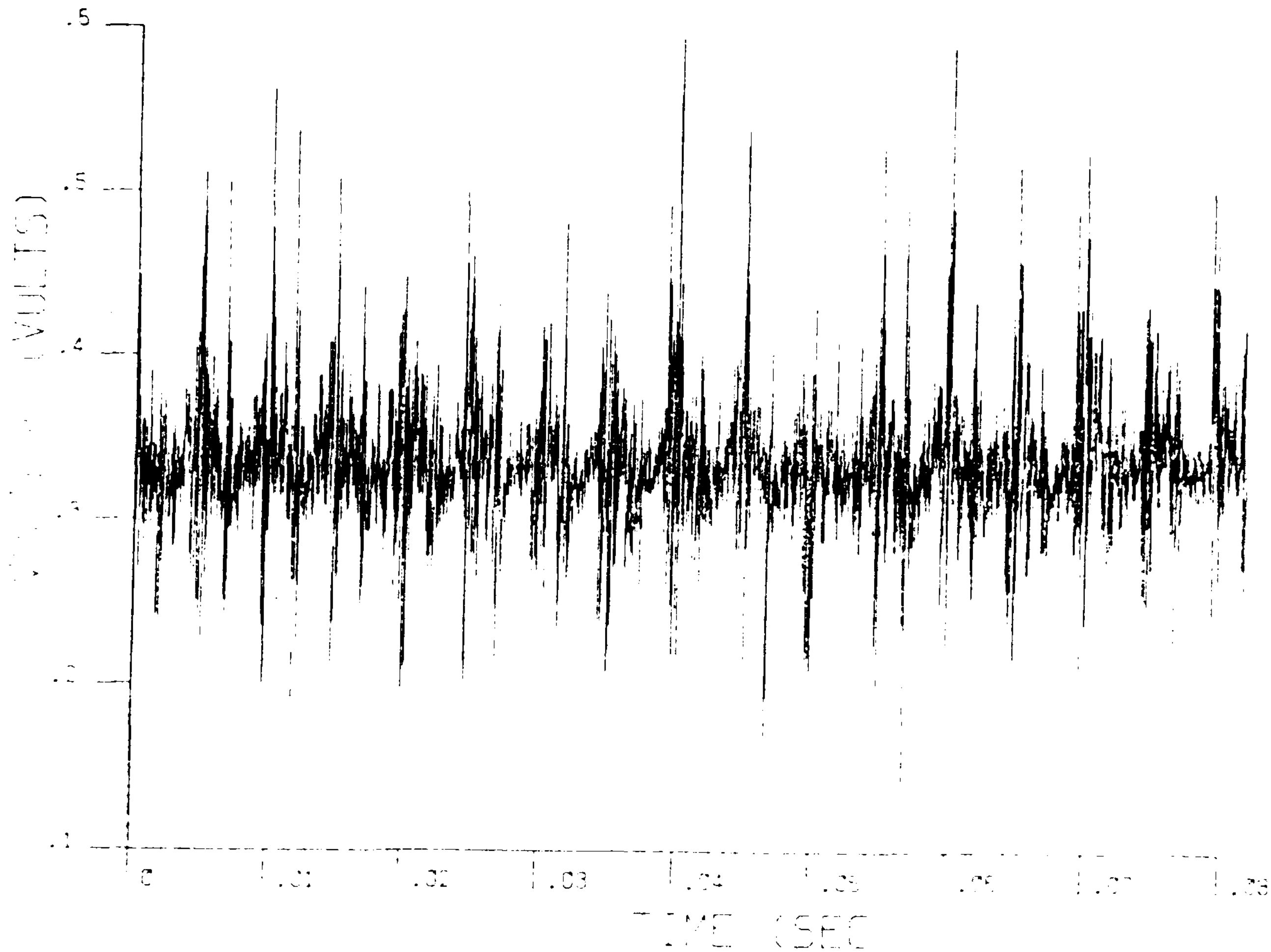


Figure 5.27 a: Time Record of Unamplified Transducer Output for Gear Pump in Undamaged State

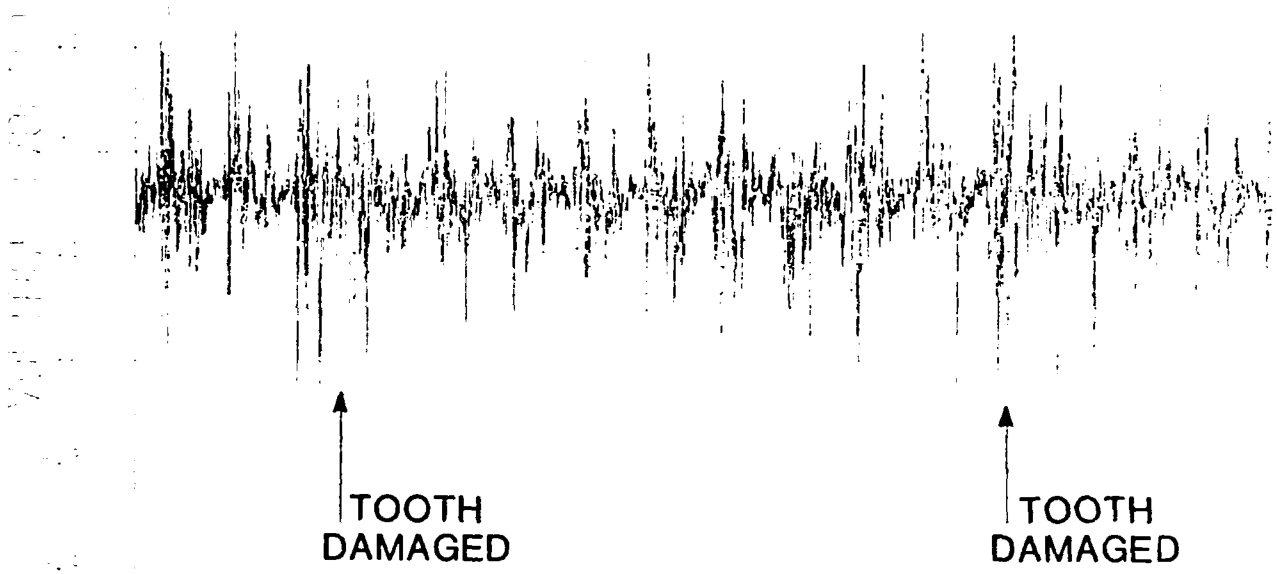


Figure 5.27 b: Time Record of Unamplified Transducer Output for Gear Pump with Gear Tooth Damage

CHAPTER VI

FIELD TESTS

6.1 Gear Pump Tests (Field Test)

The following tests were performed using the J.I. Case front-loader as a test vehicle:

(1) Pump cavitation

(2) Internal mechanical damage

6.2 Pump Cavitation

The first series of tests was designed to verify that cavitation could be detected in a pump on a field unit. The two-section gear pump on the front-loader was used for these tests. The gear pump was connected to a by-pass circuit to allow control of the inlet pressure. The test procedure involved recording the A.E. signal produced by the pump for various inlet pressures.

The AC-75L A.E. transducer was used for these experiments. The transducer was cemented to the housing of the pump, and its output was amplified using the leak detector (AVLD). The amplified signal was analyzed using the Spectral Dynamics SD-345 analyzer.

Figure 6.1 displays the spectra obtained from these tests. As can be seen from these spectra, the same general variation in A.E. signal level with inlet pressure was observed as was attained in the previous cavitation experiments. The A.E. signal initially decreases with

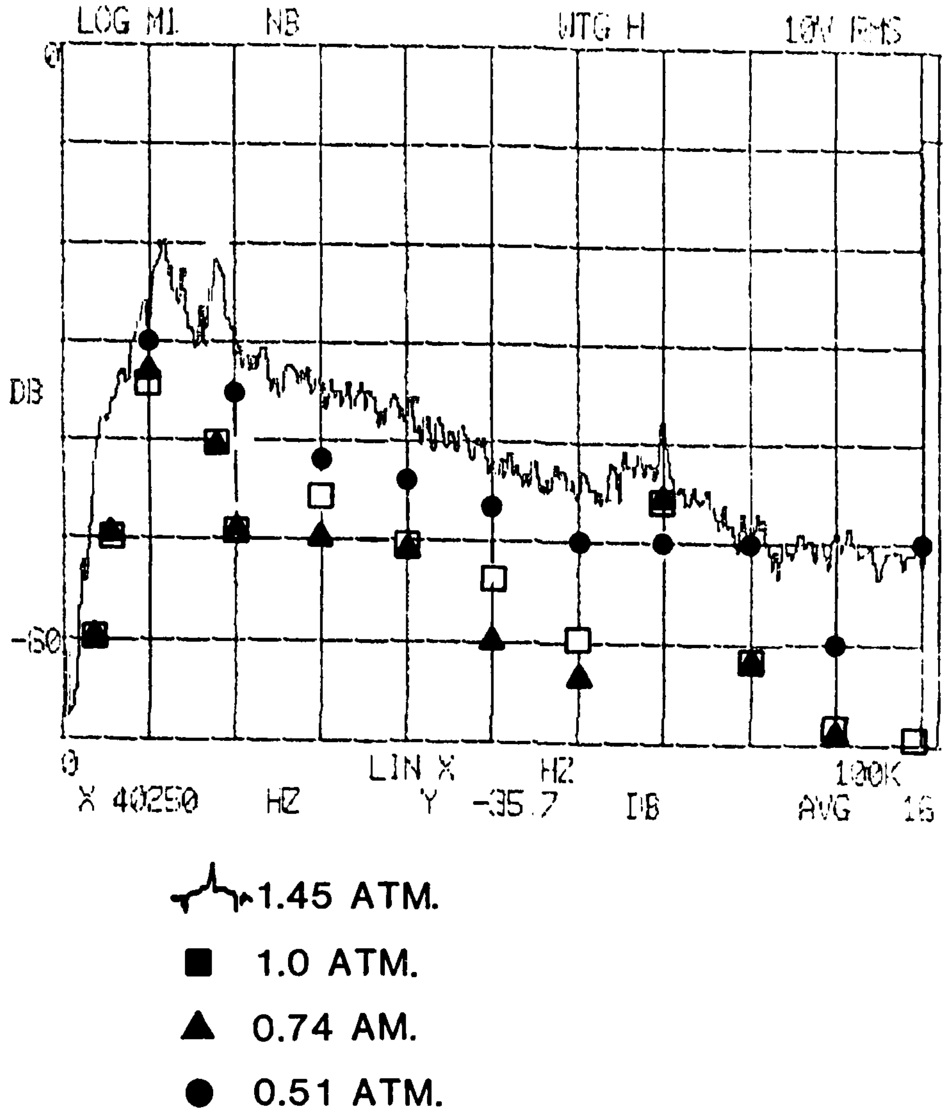


Figure 6.1: Frequency Spectra (0-100 KHz) Obtained from Front Loader Cavitation Tests

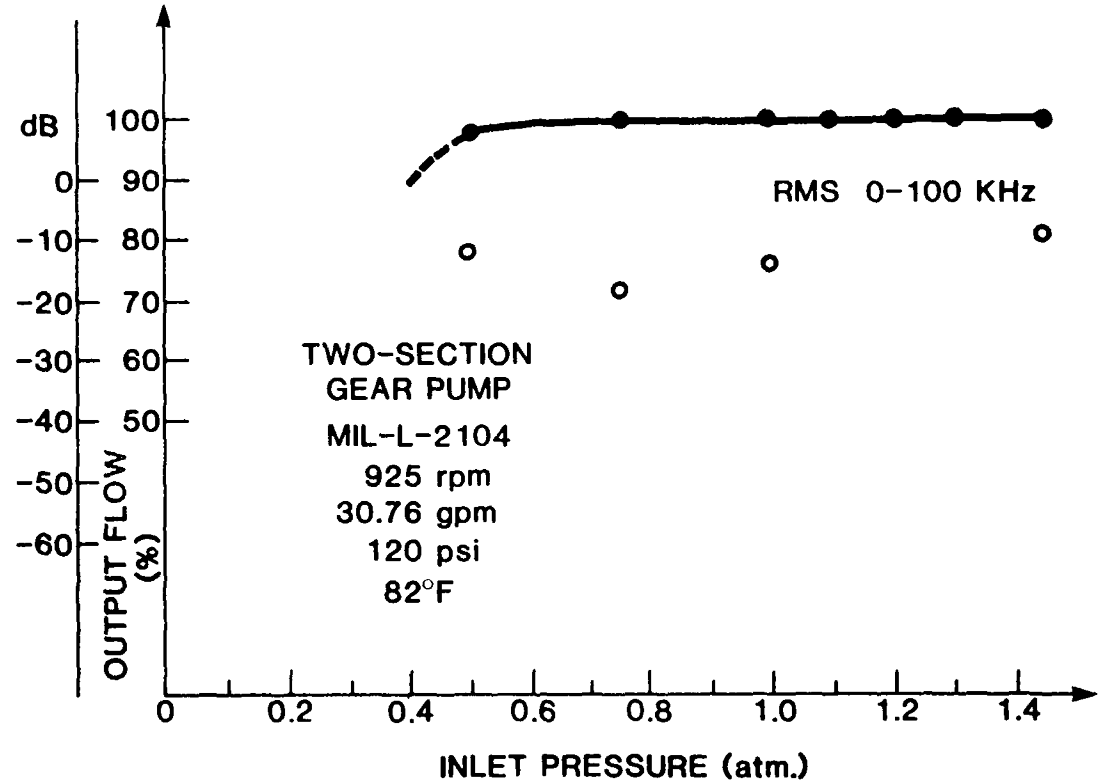


Figure 6.2: A.E. Signal Level VS. Pump Inlet Pressure Obtained from Cavitation Tests on Front Loader

decreasing inlet pressure until the point of incipient cavitation is reached. At this point, the A.E. signal level is at a minimum. Further reduction of inlet pressure at this point results in an increase in the A.E. signal level. Figure 6.2 shows this variation of the RMS of the A.E. signal with inlet pressure.

6.2.1 Mechanical Damage

A final test was performed on the two-section gear pump when installed on the loader. The purpose of this test was to verify that gear tooth damage could be detected on the loader. A pair of mating gear teeth of the pump was damaged externally by grinding. The input shaft of the pump was driven by the tractor engine motor and drive at 1200 RPM. The hydraulic circuits connected to the outlets of the pump were not loaded. This made the outlet pressure of the pump the smallest attainable.

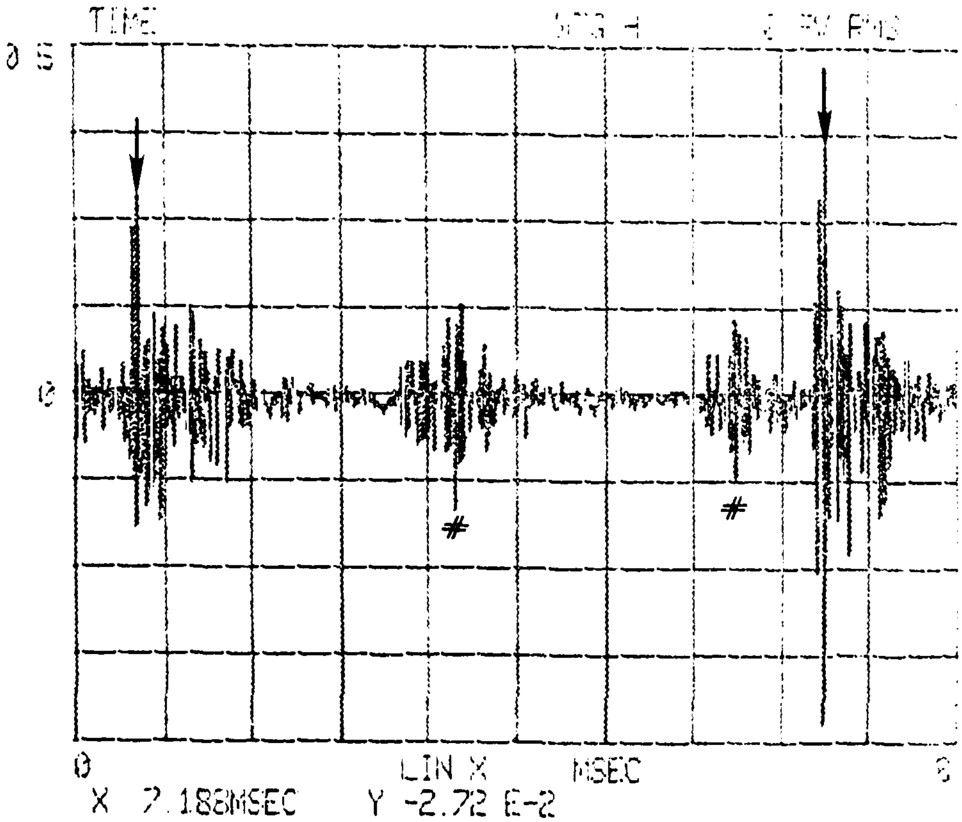
As in the test stand experiments, the frequency spectrum and time domain signal of the A.E. signal were recorded for the undamaged and the damaged states of the pump. The A.E. signal for the frequency data was amplified through the AVLD and analyzed on the spectrum analyzer. The time variation of the unamplified transducer signal was recorded using a Nicolet digital oscilloscope.

The transducer used for these tests was a Dunegan/Endevco S9204. The trandsucer was cemented to the housing of the pump over the section with the damaged gear teeth.

6.2.2 Results from Mechanical Damage Tests

Figure 6.3 contains the time variation of the unamplified transducer output. This record was obtained using the spectrum analyzer in its time mode. The record length is 8 msec and shows that individual tooth impacts are detectable from the transducer output. The two peaks denoted in the figure with arrows are two-tooth impacts from gears of the pump. The two smaller peaks between these gear tooth impacts are originated from the gear drive to the input shaft of the pump. The input shaft speed for this record was 16 Hz, which gave a gear tooth impact frequency from the pump of 160 Hz (period = 6.25 msec). The gear impact frequency from the input drive to the pump for this speed is 415 Hz.

Figures 6.4 a-b contain two frequency spectra obtained from the gear tooth damage tests on the loader. Figure 6.4-a contains the frequency spectrum obtained from the undamaged pump. Figure 6.4-b contains the frequency spectrum obtained from the pump with a severely damaged gear tooth. These spectra show no significant difference between the damaged and the undamaged states. Figures 6.5 a-b contain the time records of the unamplified transducer output. Figure 6.5-a corresponds to the undamaged pump, and Figure 6.5-b corresponds to the damaged pump. These time records clearly show that the damaged tooth meshing produces a signal significantly larger in amplitude compared to the other tooth meshings. This result shows that the amplitude of the time signal can provide information regarding the health of a gear tooth or teeth. It also suggests that bearing damage or shaft damage



DENOTES PUMP TOOTH MESHING

# DENOTES IMPUT TOOTH MESHING

Figure 6.3: Time Record of Unamplified Transducer Output for the Gear Pump on the Front Loader

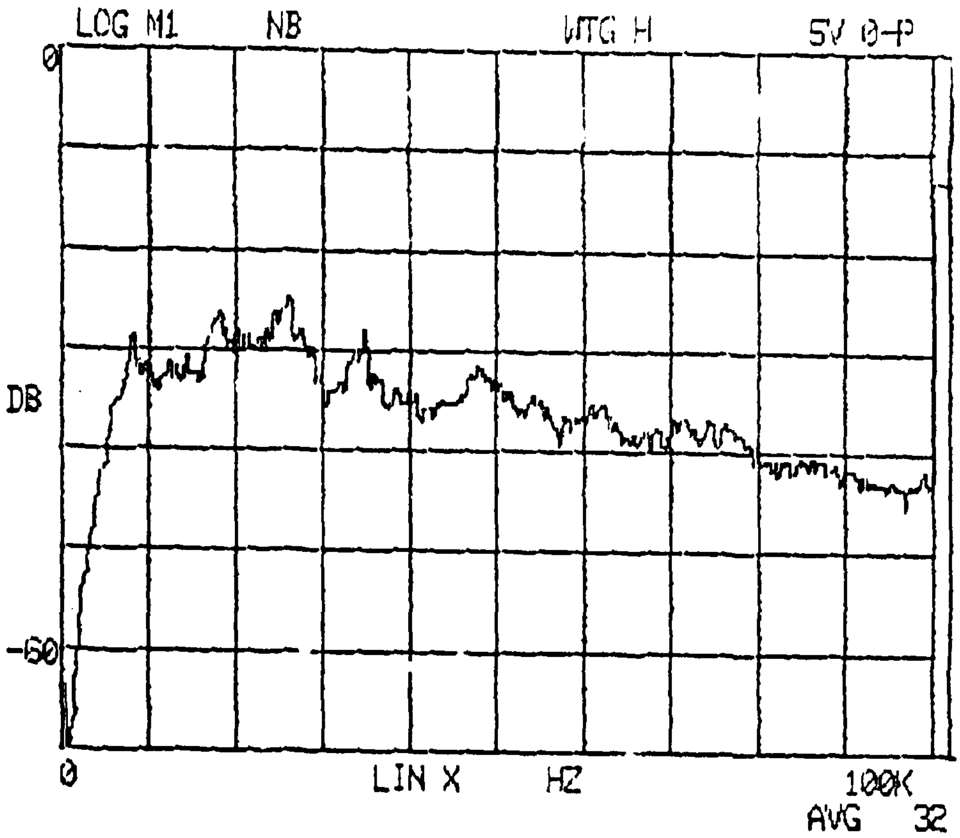


Figure 6.4 a: Frequency Spectrum of Undamaged Pump on Front Loader (0-100 KHz)

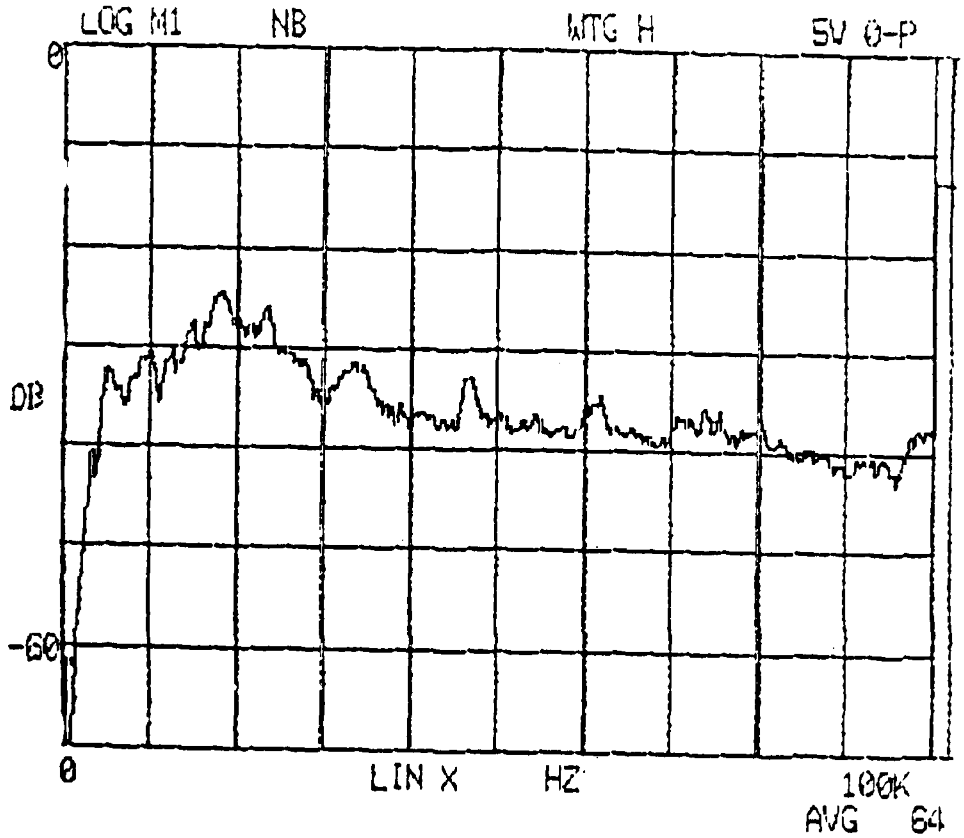


Figure 6.4 b: Frequency Spectrum of Front Loader Pump with Severe Gear Tooth Damage

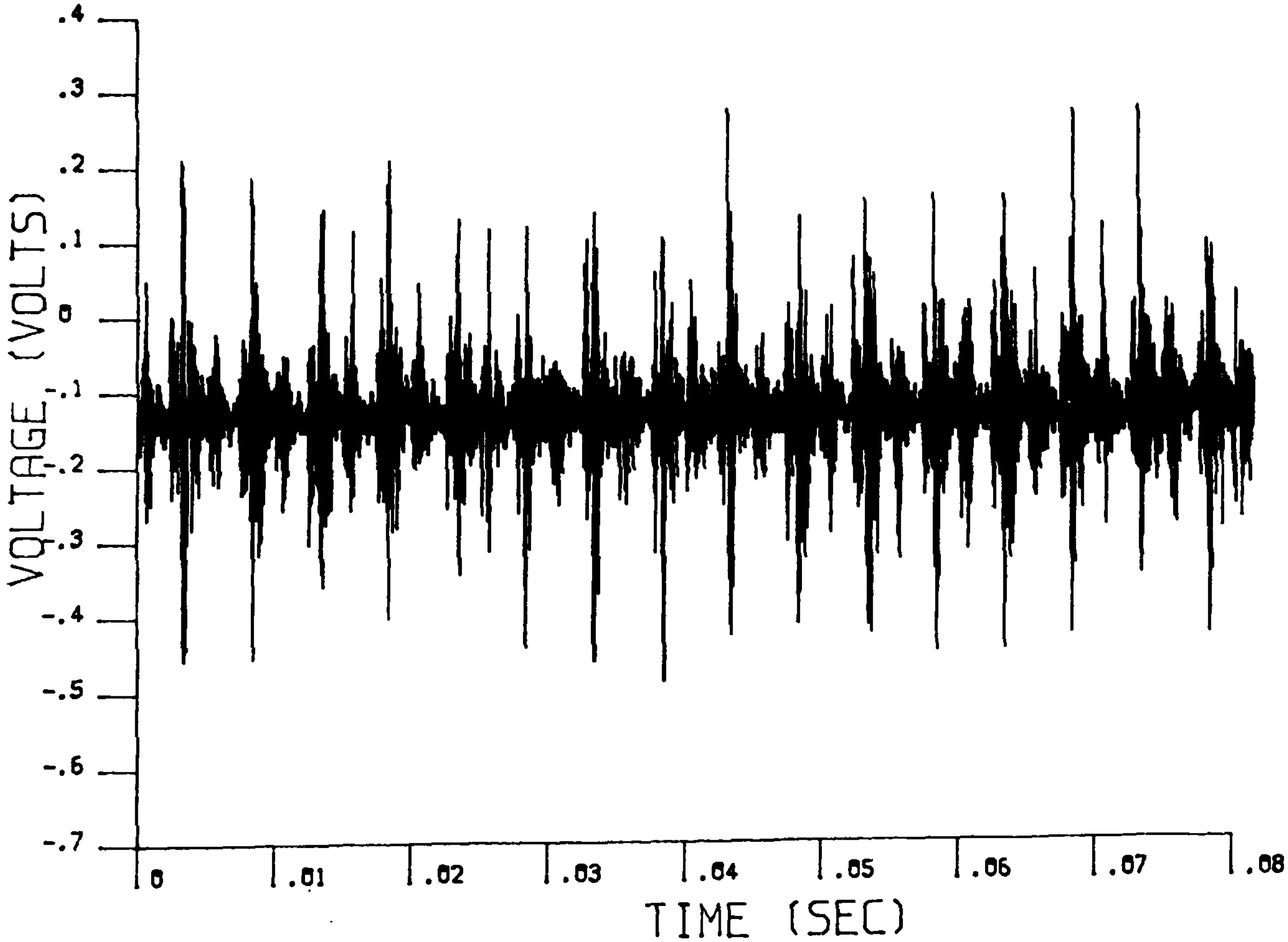


Figure 6.5 a: Time Record of Unamplified Transducer Output Obtained from Undamaged Pump on the Front Loader

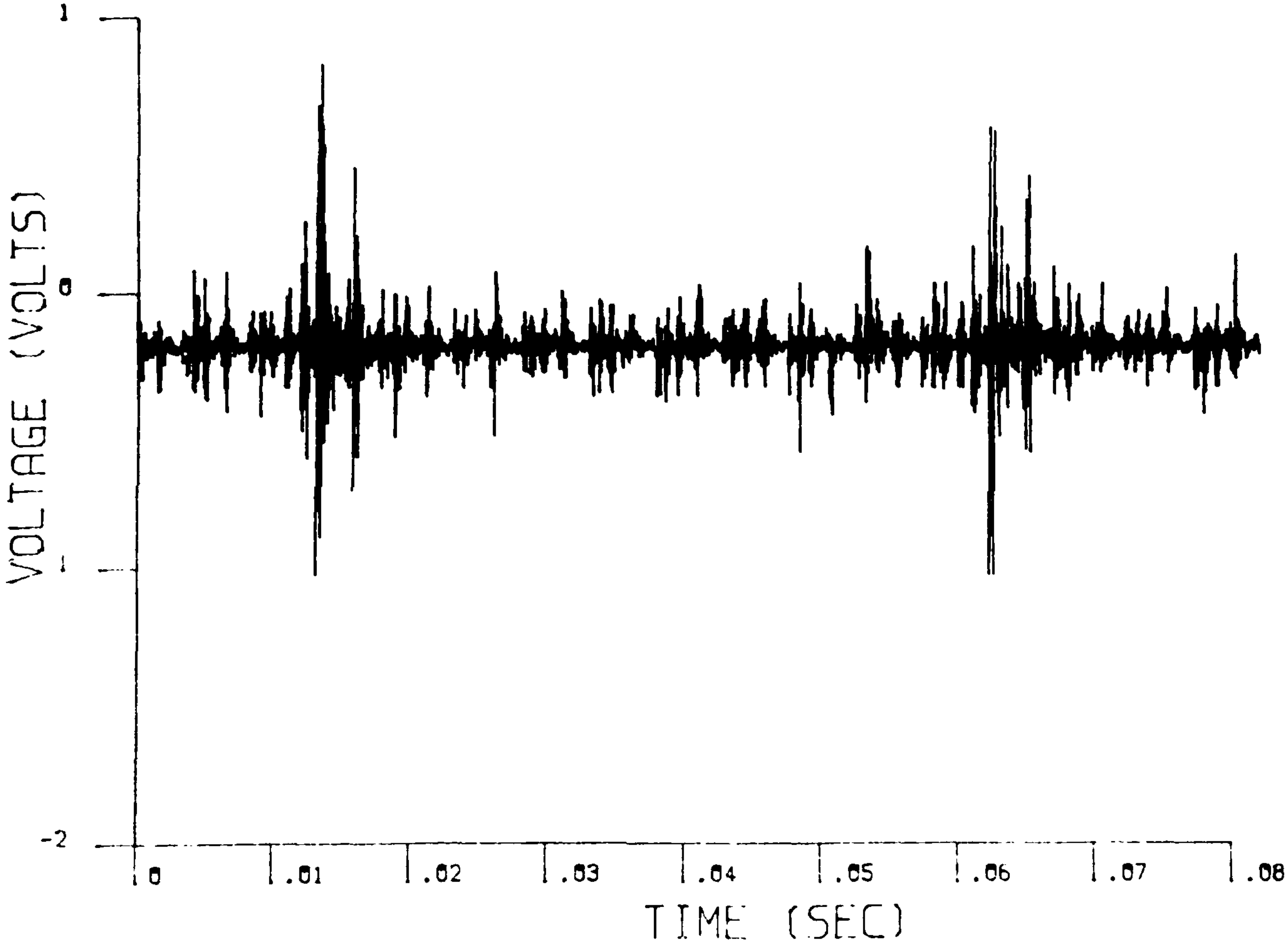


Figure 6.5 b: Time Record of Unamplified Transducer Output Obtained from Front Loader Pump with Severe Gear Tooth Damage

which causes unusual gear tooth meshings could be detected from the amplitude variation of the time signal. Though these results are only preliminary in nature, it appears the time domain signal from the transducer could provide significant diagnostic information.

CHAPTER VII

DISCUSSION AND CONCLUSIONS

Literature in the area of signature analysis, machinery health monitoring, and acoustic emission points to the idea that every type of wear, damage, or impending failure of mechanical parts creates high-frequency sound waves. Various types of leaks of gases and of liquids create high frequency sounds as well. The greatest single difficulty in implementing this kind of test procedure is that of discriminating between the signal and the normal background noise. For instance, a normal vane pump has extremely strong signal levels above 20 KHz, due to the sliding of the vanes against the housing. This sliding noise probably will mask all but the most severe bearing damage radiation. Even though severe fluid leakage around the vanes creates a strong signal, it is probably impossible to discriminate between leakage and the normal background noise.

Defects in ball and roller bearings are easily detectable in motors, spindles, and some pumps, but the signal levels may be as much as 60 dB below (say) the signal level of a badly worn tooth in a gear pump. If, however, a bearing becomes worn to the extent that misalignment of the gears or vanes causes interference with the housing, the change in signal output should be readily measurable.

Most of the tests in this study involved using Fast Fourier Transform analysis or ordinary time domain analysis. It is likely that more refined techniques involving cross-correlation, cross power spectral density, envelope detection, and cepstrum analysis would yield good results on the weaker signals.

The results of this study show that Acoustic Emission instrumentation can detect worn or damaged hydraulic components that are still functional. Thus, the procedure described here would be capable of detecting impending failure in many types of machines.

The specific conclusions are as follows:

(1) Leakage rates of approximately 10 mL/min. can be detected in valves and cylinders if hydrostatic pressurization is used. These signal levels are so low that they are obscured by normal pump noise in an operating system. Frequency domain representation is the best way to test for leakage. These procedures should be translatable to any kind of system where liquid leaks are concerned.

(2) Normal gradual wear in gear pumps and vane pumps is difficult to detect unless the signal is compared with a baseline signal maintained for each component.

(3) Defects such as damaged gear teeth or roller bearings can be detected in gear pumps. Time-domain data are easier to interpret than Fast Fourier Transform data.

(4) The best test condition for finding damaged parts in hydraulic pumps is full flow and zero outlet pressure.

(5) Transducer coupling is very critical, since a change in acoustic impedance at the interface can cause signal strength variations of up to 10 dB. Whenever possible, the transducer should be cemented to a flat-ground test spot or used with a spring loaded adaptor.

(6) The "leak-detector" type instrument with a range of 10 KHz to 100 KHz is the most useful system for detecting damaged parts in hydraulic systems. The addition of an output from the amplifier section before the signal is heterodyned makes the operation more flexible, in that the raw signal can be fed directly to an FFT analyzer.

(7) Transducers with high sensitivities and resonances between 30 KHz and 100 KHz are the most useful for monitoring hydraulic components.

(8) Signals from gear-tooth wear or damage may be so strong that they can be fed directly to an oscilloscope in some applications. Of course, care must be taken that the high frequency signals are not lost due to the impedance mismatch.

(9) Thrust-plate leakage in operating gear pumps is difficult to detect, since it tends to be masked by mechanical noise.

(10) Defects, wear, and contamination can be detected in ball bearings if background noise levels are not too high.

(11) Incipient cavitation in pumps can be readily detected by Acoustic Emission transducers.

In terms of the main objective, the overriding conclusion is that Acoustic Emission techniques can be used to detect damage in many, but not all, of the hydraulic components used in earth moving equipment.

CHAPTER VIII

SUGGESTIONS FOR FUTURE WORK

(1) Leakage phenomena in valves and cylinders should be investigated in greater depth. The effects of leakage path or orifice geometry should be considered.

(2) More sophisticated signal processing techniques, such as envelope detection and correlation, cepstrum analysis, and correlation computation should be explored

(3) The acoustic emission properties of systems uitlizing water-based fluids should be studied.

(4) Refinement of A.E. transducers for field application should be pursued. In many situations, a larger, more sensitive, transducer having a broad resonance of around 100 KHz might be more suitable for leak detection.

(5) The application of the above-stated signal processing techniques to the detection of internal mechanical damage, such as shaft damage, bearing, and pressure plate damage should be explored.

APPENDIX A

REFERENCES

REFERENCES

1. Parsons, E. U., "Acoustic Emission Detects Failures," Physical Acoustics Corporation, Technical Report TR-20, July 1975.

2. Block, H. P. and R. W. Finley, "Using Modified Acoustic Emission Techniques for Machinery Condition Surveillance," Presented at Seventh Turbo Machinery Symposium, Houston, Texas, Dec., 1978, Gas Turbine Laboratories, Texas A&M University, College Station.

3. Drago, R. J. and D. B. Board, "High Frequency Vibration Monitoring Techniques for Gear/Bearing-System Failure Detection," AGMA Paper No. 109.36, October 1975.

4. Block, H. P., "Development and Experience with Computerized Acoustic Incipient Failure Detection (IFD) Systems," ASME Paper, Presented at the Energy Technology Conference & Exhibit, Houston, Texas, September 1977.

5. Balderston, H. L., "The Detection of Incipient Failure of Bearings," Presented at the 28th National Fall Conference of the American Society for Nondestructive Testing, October 14-17, 1968, Detroit, Michigan.

6. Love, A.E.H., "Treatise on the Mathematical Theory of Elasticity," 1944, p. 284-286, 451-453, Dover Publications Inc., New York.

7. Block, H. P., "Predict Problems with Acoustic Incipient Failure Detection Systems," Hydrocarbon Processing, October 1977.

8. Block, H. P., "Acoustic Incipient-Failure Detection," The Oil and Gas Journal. February 6, 1978.

9. Finley, R. W., "Failure Detection in Machinery," Chemical Engineering, July 14, 1980.

10. Dickey, J., J. Dimmick, and P. M. Moore. "Acoustic Measurement of Valve Leakage Rates," Materials Evaluation, January 1978.

11. Borst, J. F., "Acoustic Emission - Is It a Promise or Mirage," Nuclear Engineering International, May 1977, pp. 54-56.

12. Manual on "5120 Acoustic Valve Leak Detector." Physical Acoustics Corporation, Princeton, New Jersey.

13. Manual, D. N., "Portable Activity Monitor Model 4103," Dunegan/ Endevco, California.

14. Manual on "6100 Bearing Monitor," Physical Acoustics Corporation, Princeton, New Jersey.

15. Manual on "SD345 Model III Spectrascope," Spectral Dynamics.

16. Subramaniam, K. V., "Acoustic Measurement of Hydraulic Cylinder Leakage Rates," The BFPR Journal, 1984, Volume 17, No. 2.

APPENDIX B

SELECTED BIBLIOGRAPHY

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1. Igarashi, Teruoi, and Hiroyoshi Hamada, "Studies on the Vibration and Sound of Defective Rolling Bearings - I. Vibration of Ball Bearings with One Defect," Bull JSME, Vol. 15, No. 204, June 1982, pp. 994-1001.

2. Chaturvedi, G. K., and D. W. Thomas, "Bearing Fault Detection Using Adaptive Noise Cancelling," ASME Paper 81-DET-7.

3. Osuagwu, C. C., and D. W. Thomas, "Effect of Inter-Modulation and Quasi-Periodic Instability in the Diagnosis of Rolling Element Incipient Defect," ASME Paper 81-DET-11.

4. Eshleman, Ronald L., "Role of Sum and Difference Frequencies in Rotating Machinery Fault Diagnosis," I. Mech. E. Conf. Publ. 1980-4, 2nd - Cambridge, England, Sept. 1-4, 1980. Published by Mech. Eng. Publ. Ltd., Bury St., Edmunds, Suffolk, England Inst. of Mech. Eng., London, England, 1980, pp. 145-149.

5. Burchill, R. F., and J. L. Frarey, "Pump Diagnostics Through Vibration Analysis," Proc. Natl. Conference Fluid Power Annual Meet, 35th, Vol. 33, Chicago, IL., Nov. 13-15, 1979.

6. Mitchell, John S., "Bearing Diagnostics: An Overview," Diagn. Machine Health, Symposium presented at Winter Annual Meeting, San Francisco, CA, Dec. 10-15, 1978. Published by ASME, San Francisco, CA, 1978, pp. 15-24.

7. Hall, Godfrey M., "Diagnose Vibration Problems by Applied Spectrum Analysis," Pulp. Paper Can., Vol. 79, No. 6, June 1978, pp. 97-99.

8. Braun, S., and B. Datner, "Analysis of Roller/Ball Bearing Vibrations," ASME Paper 77-WA/DE-5 for Meeting Nov. 27-Dec. 2, 1977.

9. Dyer, D., and R. M. Stewart, "Detection of Rolling Element Bearing Damage by Statistical Vibration Analysis," ASME Paper 77-DET-83 for Meeting, Sept. 26-30, 1977.

10. Winn, L. W., and H. L. Bull, "Diagnostic System for Ball Bearing Quality Control," SAE Paper 760910 for Meeting Nov. 29-Dec. 2, 1976.

11. George, J. A., T. C. Mayer, and E. F. Covill, "Evaluation of Shock Pulse Diagnostic Technique to the UH-1 Series Helicopter," Shock Vib. Bull, No. 45 1974-1975, Proc. 45th Symposium on Shock and Vibration, Dayton, Ohio, Oct. 22-25, 1974.

12. Houser, Donald R., "Signal Analysis Techniques for Vibration Diagnostics," Natl. Bureau of Standards Spec. Publ., No. 436, 1975, for 22nd Meet of Mech. Failures Prev. Group, Apr. 23-25, 1975, pp. 3-17.

13. Shives, T. Robert, and William A. Willard, (eds.), "Detection, Diagnosis, and Prognosis," Natl. Bureau of Standards Stand Special Publication, No. 436, 1975, for 22nd Meeting of Mechnical Failures Prev. Group, Anaheim, CA, Apr. 23-25, 1975.

14. Sparks, Cecil R., and J. C. Wachel, "Quantitative Signature Analysis for On-Stream Diagnosis of Machine Response," Mater. Eval., Vol. 31, No. 4, April 1973.

15. Borhaus, Jan E., and John S. Mitchell, "Widened Frequency Range is Improving Today's Machinery Vibration Analysis," Power, Vol. 117, No. 3, March 1973, pp. 51-53.

16. Weichtbrodt, B., and K. A. Smith, "Signature Analysis Non-Destructive Techniques for Incipient Failure Identification Application to Bearings and Gears," MBS Spec. Publ. 336, Proc. 5th Space Simulation Symp. Conf., pp. 407-48.

17. Gersham, S. G., and V. I. Povarkov, "Investigation of Spectral and Correlation Characteristics of Ball Bearing Vibrations," Int. Congress on Acoustics, 6th, Tokyo, Vol. 4, Aug. 21-28, 1968, Paper F-5-17, pp. 197-200.

18. Board, David B., "Incipient Failure Detection in High-Speed Rotating Machinery," Symp. on Nondestructive Evaluation. 10th Proc., San Antonio, TX, April 23-25, 1975, pp. 8-18.

19. Braun, S. G., and B. B. Seth, "Signature Analysis Methods and Applications for Rotating Machines," ASME Paper 77-WA/Aut-5 for Meeting, Nov. 27-Dec. 2, 1977.

20. Braun, Simon G., "Acoustic Signature Analysis and Time Domain Techniques for Detecting Flaws in Rotating Machinery," Ultrasonic Symposium Proceedings, Sept. 25-27, 1978.

21. Rutter, Thomas A., "Acoustic Analysis of Quiet Ball Bearing Failure Modes," Mar. Technology, Vol. 16, No. 2, April 1979, pp. 181-188.

22. Rogers, L. M., "Application of Vibration Signature Analysis and Acoustic Emission Source Location to On-Line Condition Monitoring of Anti-Friction Bearings," Tribol. Int., Vol. 12, No. 2. April 1979, pp. 51-59.

APPENDIX C

ABSTRACTS OF RELATED PAPERS

ABSTRACTS OF RELATED PAPERS

Machinery Health Monitoring:

EARLY BEARING FAILURE DETECTION BY SPIKE ENERGY MEASUREMENT
Braithwaite, K. G.
IRD Mechanalysis Ltd, Stoney Creek, Ont., Canada
Pulp Pap Can., Vol. 82, No. 11, Nov. 1981, p. 101-104.

Background and inherent low frequency vibrations can often overshadow high frequency failing anti-friction bearing vibration when using the conventional parameters of displacement, velocity or acceleration. Spike energy measurements (g-SE) taken with hand-held portable instrument or a permanent machine condition monitor can overcome these shortcomings. Trending and conventional analysis techniques, together with expected frequencies, are discussed in conjunction with common anti-friction bearing problems.

ACOUSTIC SIGNATURE ANALYSIS AND TIME DOMAIN TECHNIQUES FOR DETECTING FLAWS IN ROTATING MACHINERY
Braun, Simon G.
Ford Motor Co., Detroit, Mich.
Ultrason Symp. Proc., Cherry Hill, NJ, Sept. 25-27, 1978.
Published by IEEE, New York, NY, 1978, p. 277-285.

A general approach to diagnostic oriented signature analysis is presented dealing with available model of signal generation and transmission path, instrumentation, data reduction and transformation. noise mechanisms, and feature extraction. The case of rotating machinery is specific by virtue of the signal's general behavior being dictated by the rotational frequency. The well established technique of spectrum analysis is extremely sensitive to more elaborate techniques, and some new time-domain approaches are presented.

SIGNATURE ANALYSIS METHODS AND APPLICATIONS FOR ROTATING MACHINES
Braun, S. G.; Seth, B. B.
Ford Motor Co., Detroit, Mich.
ASME Paper No. 77-WA/Aut-5 for Meet Nov. 27-Dec. 2, 1977, p. 8

This paper describes the techniques of decomposition of signals obtained from a rotating machine. Time-domain averaging and variance techniques have been described, along with examples illustrating the extraction of periodic and repetitive nonperiodic signals. A discussion of other methods of signature analysis is also included. A technique for developing ideal digital filters by modifying the discrete Fourier tranform coefficients is also described. Illustrated are cases of high pass and multiple narrow-band pass filtering applied to specific measurement cases.

INCIPIENT FAILURE DETECTION IN HIGH-SPEED ROTATING MACHINERY

Board, David B.
Boeing Vertol Co., Philadelphia, PA
Symp. on Nondestr. Eval., 10th Proc., San Antonio, TX, Apr. 23-25, 1975, p. 8-18.

A new technique for high frequency vibro-acoustic emission analysis was evaluated on three CH-47 drive system transmissions in a regenerative test stand. Test results indicate that this new technique for high frequency vibra-acoustic analysis shows excellent potential for early stage "IN-SITU" detection and identification of faults in complex, high-speed, rotating machinery.

INCIPIENT FAILURE DETECTION

Drago, Raymond J.
Boeing Vertol Co., Philadelphia, PA.
Power Transm. Des., Vol. 21, No. 2, Feb. 1979, p. 40-45.

Incipient Failure Detection (IFD) is a new technique for detecting the presence and monitoring the progression of faults in rotating machinery. IFD uses inexpensive sensors and does not require custom baseline data. Its operation is explained and some typical test results are presented. Incipient failure detection is a specialized form of high-frequency vibration analysis. The basic premise of IFD is that dynamic events related to defects in rotating machinery will cause vibration amplitude modulations in the time domain across a broad frequency spectrum. A bandpass filter can monitor amplitude variation of a narrow band of energy at a low-noise carrier frequency while screening out energy variations in the rest of the frequency spectrum. Spectrum analysis of the envelope-detected signal will show a strong peak at the defect frequency but a relatively flat response at all other frequencies.

APPLICATION OF VIBRATION SIGNATURE ANALYSIS AND ACOUSTIC EMISSION SOURCE LOCATION TO ON-LINE CONDITION MONITORING OF ANTI-FRICTION BEARINGS
Rogers, L. M.
Unit Insp. Co., Sketty Hall, Swansea, Wales
Tribol. Int., Vol. 12, No. 2, April 1979, p. 51-59.

This paper describes the slow-speed application of acoustic emission to in-service monitoring of the integrity of offshore production platform slewing cranes and the high-speed application of Kurtosis to monitoring the condition of rolling element bearings in medium to fast rotating machinery.

ACOUSTIC ANALYSIS OF QUIET BALL BEARING FAILURE MODES
Rutter, Thomas A.
Puget Sound Nav. Shipyard, Bremerton, WA.
Mar. Technology, Vol. 16, No. 2, April 1979, p. 181-188.

Qualitative analysis of rotating machinery structure-borne noise and vibration can identify ball bearing installation problems. Indications of improper lubrication and impending bearing failures are identified. The problem of nonuniform motion of ball bearing components in determining expected bearing frequencies is noted together with the effects of grease deterioration. The philosophy of structure-borne noise versus vibration measurements and an appreciation of several analysis techniques in the context of determining maintenance requirements and evaluating subsequent repair are discussed.

NEW MACHINERY HEALTH DIAGNOSTIC TECHNIQUES USING HIGH-FREQUENCY VIBRATION
Burchill, R. F.; J. L. Frarey; and D. S. Wilson
Shaker Res. Corp.
SAE Prepr. Pap. No. 730930 for Meet Oct. 16-18, 1973, p. 8 CODEN: SEPPA8.

A technique is discussed for generations diagnostic information from the vibration signature of machinery in the high-frequency range (up to 100 KHz). The signal generation mechanism is discussed, as well as the diagnostically significant characteristics of the data and a method of extracting this information. Two specific cases are presented utilizing the technique to illustrate its suitability for many of the common problems encountered in machinery.

BOATING MACHINERY DIAGNOSIS THROUGH SHOCK PULSE MONITORING
Board, David
SKF Ind. Inc., Kins of Prussia, PA.
Diagn. Mach. Health, Symp. Presented at Winter Annual Meet. of ASME, San Francisco Calif., Dec. 10-15, 1978, Publ. by ASME, San Francisco, CA., 1978, p. 25-40.

During recent years, vibration analysis has been receiving renewed emphasis as a means of monitoring rotating machinery for discrepant operating and mechanical conditions. It has been through this recent research, that two principle methods of vibration analysis have evolved (See References). These two methods are frequently referred to as low frequency vibration analysis (or pattern recognition) and high frequency vibration analysis (or shock pulse monitoring). This paper will discuss shock pulse theory, illustrate its use through case histories, and describe new automated hardware that is simple to operate and maintain in the field, as well as a cost effective means of preventing catastrophic failure or unforseen equipment downtime.

Hydraulic Systems;

ACOUSTIC MEASUREMENT OF VALVE LEAKAGE RATES
Dickey, Joseph; Joseph Dimmick, and Paul M. Moore
David W. Taylor Naval Ship Res. & Dev. Center, Annapolis, MD.
Mater. Eval., Vol. 36, No. 1, Jan. 1978, p. 67-77.

The acoustic emission associated with leakage through air, steam, hydraulic, and water valves was investigated. The experiments were designed to determine what characteristics of the acoustic emission may be related to leak rate with the hope that instrumentation could be developed to acoustically detect and measure leakage. It is concluded that, in the air and steam valves tested, the amplitude of the acoustic emission at certain frequencies was an indicator of leak rate.

ACOUSTICAL VALVE LEAK DETECTOR FOR FLUID SYSTEM MAINTENANCE
Dimmick, Joseph G.; Jack R. Nicholas; and Joseph Dickey; and Paul M. Moore.
David W. Taylor Nav. Ship Res. & Dev. Center, Annapolis, Md.

A portable, passive, non-destructive, non-intrusive instrument is currently in use by the U.S. Navy to detect fluid leakage through shipboard steam, water, hydraulic, and high-pressure air valves. The

Acoustic Valve Leak Detector (AVLD) was developed by David W. Taylor Naval Ship R & D Center. The instrument is used to identify more precisely internal leakage points which may require opening and visual inspection of submarine piping systems. "Open-and inspect" routines generate both risk of system damage and high man-hour expenditure. The AVLD is currently being used for troubleshooting, for overhaul planning, and in a systematic preventive maintenance program for seawater valves. Plans for future development and progress achieved in current work are described.

PUMP DIAGNOSTICS THROUGH VIBRATION ANALYSIS
Burchill, R. F., and J. L. Frarey
Shaker Res. Corp., Ballston Lake, NY.
Proc. Natl. Conf. Fluid Power Annual Meet. 35th, Vol. 33, Chicago, IL, Nov. 13-15, 1979. Sponsored by Illinois Inst. of Technology, Chicago, 1979. p. 97-100. CODEN:PDFPAD

Methods used to diagnose machine problems prior to failure are reviewed. Early detection of machinery problems can be accomplished using high frequency acceleration in the frequency range to 100 KHz. High frequency analysis techniques when applied to pumps can be helpful in determining seal and impeller rubs, bearing wear and damage and the onset of cavitation. This type of information can be detected in advance of performance deterioration to provide the user with early indications of pump degradation.

APPENDIX D

PUMP SPECIFICATIONS

PUMP SPECIFICATIONS

Vane Pumps- Sperry-Vickers, Model VIO-15-75

Capacity cu.in./RV	1.34
Maximum Pressure (psi)	2000
Maximum Speed (RPM)	1800
Viscosity Range	20-600
Maximum Temperature ^{o}F	180
Minimum Speed (RPM)	600
Number of Vanes	12

Piston Pump- Hydro-Rene Leduc, Model PB527

Capacity cu.in./RV	2.31
Maximum Pressure (psi)	5500
Maximum Speed (RPM)	2500
Viscosity Range (CSE)	20-600
Maximum Temperature ^{o}F	180
Minimum Speed (RPM)	400
Number of Pistons	5

One-Section Gear Pump- Commercial Shearing,Inc., Model P25X342 AB15-25

Capacity cu.in./RV	4.53
Maximum Pressure (psi)	3000
Maximum Speed (RPM)	3000
Viscosity Range (CSE)	20-600
Maximum Temperature °F	180
Minimum Speed (RPM)	400
Number of Teeth	10

Two-Section Gear Pump- Commercial Shearing, Inc., Model PSOB299BI 0520

	Section I	Section II
Capacity cu.in./RV	3.46	6.65
Maximum Pressure (psi)	2500	2500
Maximum Speed (RPM)	2400	2400
Viscosity Range (CSE)	20-600	20-600
Maximum Temperature ^{o}F	180	180
Minimum Speed (RPM)	600	600
Number of Teeth	10	10

APPENDIX E

NOTES ON FREQUENCY ANALYSIS

NOTES ON FREQUENCY ANALYSIS

The analyzer discussed in Chapter III gives a "real-time" representation, meaning that for certain frequency ranges, no data is lost. As the internal micro-processor digitally integrates the data, the incoming data is stored. Conventional analog analyzers cannot do this since they are "looking" through a moving "window" and the other data is being discarded.

The FFT analyzer, however, has a shortcoming in that the entire frequency range of 100 KHz is covered by only four hundred vertical lines giving a "window" width, or resolution, of 2500 Hz. The analog type analyzer can be adjusted from 1 Hz to as broad as needed. Therefore, if the signal band of interest was only 10 KHz or so in width, an analog analyzer, or even a fixed filter could give much better resolution at lower cost. It would not be practical to use the analog analyzer over a wide range with a narrow "window" because the scanning time would be prohibitively large.

It seems probable, as a result of this study that a simple fixed analog filter with a transducer, preamplifier and root-mean-square volt meter would be sufficient for those tests, outlined in the body of this report, where the level frequency band is a good criterion. In applications such as detecting damaged gear teeth or vanes, an oscilloscope would be needed to monitor the output of the filter.